ARTI-21CR/605-50010-01-Pt. 2

PROPERTIES AND CYCLE PERFORMANCE OF REFRIGERANT BLENDS OPERATING NEAR AND ABOVE THE REFRIGERANT CRITTCAL POINT

Task 2: **Air** Conditioner **System** Study

Final Report

Date Published – September 2002

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Use of Non-SI Units in a Non-NIST Publication

It is the policy of the National Institute of Standards and Technology to use the International System of Units (metric units) in all of its publications. However, in North America in the HVAC&R industry, certain non-SI units are so widely used instead of SI units that it is more practical and less confusing to include measurement values for customary units only in figures and tables describing system performance.

EXECUTIVE SUMMARY

The main goal of this study was to investigate performance of an **R410A** air conditioner relative to an **R22** air conditioner, with specific interest in performance at high ambient temperatures at which the condenser of the R410A system may operate above the critical point. The study comprised experimental and modeling efforts.

Within the experimental part of the study we tested split system 3-ton **R22** and **R410A** residential air conditioners. The selected systems comprised identical evaporators and condensers, respectively, and were equipped with thermostatic expansion valves (TXVs). We tested the **R22** air conditioner in the 82.0 "F to 135.0 "F **(27.8** "C to **57.2** "C) outdoor temperature range. We planned the same range of ambient temperatures for the **R410A** system, however, the **R410A** compressor's safety system cut off the compressor at **135.0 °F (57.2 °C**) outdoor temperature, and the **130.0** "F **(54.4 "C)** test was the highest temperature at which measurements were taken with the original R410A compressor. Subsequently, a custom-manufactured **R410A** compressor **was** installed in which the safety system was disabled and the electric motor was more powerful than in the original compressor. With **this** new compressor, we took data at up to **155.0** "F (68.3 "**C)** ambient temperature, at which the system operated in a transcritical mode.

The **R22** and **R410A** systems operated normally during all tests, and visual observations of the **R410A** system provided no indication of vibrations or **TXV** hunting at high ambient temperatures with compressor discharge in the transcritical regime. The tests showed that capacity and energy efficiency ratio **(EER)** for both systems were nearly

linearly dependent on the ambient temperature, with the performance degradation of the R410A air conditioner being greater than that for the R22 air conditioner. The R22 and R410A systems had a similar capacity and EER at the 82.0 °F (27.8 °C) rating point, but the R22 air conditioner was a better performing system at higher temperatures. The report contains a description of the test facility, test procedures, and detailed test results.

The modeling part of this project provided a thrust for the final stage of preparing a beta version of EVAP-COND, a windows-based simulation package for predicting performance of finned-tube evaporators and condensers. Both the evaporator and condenser models can account for one-dimensional non-uniform air distributions and interaction between the air and refrigerant distributions. The visual interface helps with specifying tube-by-tube refrigerant circuitry and analyzing detailed simulation results on a tube-by-tube basis. This feature facilitates designing optimized heat exchangers. Ten refrigerant and refrigerant mixtures are available. EVAP-COND uses REFPROP6 (McLinden et al., 1998) routines for calculating refrigerant properties.

The modeling part of this study also included formulation of a model for a TXV-equipped air-conditioner, which was then used to simulate performance of R22, R410A, R404A, and R134a air conditioners. The model uses the EVAP-COND evaporator and condenser models, and simulates the compressor using a compressor map algorithm. The same as for EVAP-COND, the air conditioner model is REFPROP6-compatible and technically can include any refrigerant and refrigerant mixture that is covered by REPFROP6. We validated the system model and performed simulations for the four refrigerants **for** the **82.0 °F** to **135.0 °F (27.8 °C** to **57.2 °C)** outdoor temperature range using the same heat exchangers that were tested with R22 and R410A systems. The simulations results are consistent with the test results obtained for **R22** and **R410A** and can be explained in terms of refrigerant thermophysical properties and their impact on performance in a system with non-optimized heat exchangers.

ACKNOWLEDGEMENT

This work was sponsored by the Air-conditioning and Refrigerantion Technology Institute under ARTI 21-CR Program Contract Number 605-50010. Supplementary funding was provided by the US. Department of Energy, Contract Number DE-AI01-97EE23775, and NIST. We acknowledge the feedback and support from people associated with the sponsoring organizations, including Michael Blanford, Mark Spatz, Barbara H. Minor, Lawrence R. Grzyll, Dick Ernst, Steven Szymurski, and Esher Kweller. The two tested systems were contributed by Lennox Industries Inc. Len Van Essen provided essential advice during the unit selection process. Copeland Corporation contributed two custom-fabricated R410A compressors for transcritical tests of the R410A system along with valuable cooperation of Ken Monnier, Jim Horn, and Stan Schumann. John Wamsley provided laboratory support needed for tests in the NIST environmental chambers, and Samuel Y. Motta and Yongchan Kim assisted with their technical comments and cooperation. The authors also thank Mark O. McLinden and Keith Rice, principal investigators of the other two tasks associated with this project, for their interactions and comments.

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NOMENCLATURE

A_f = finned surface area (ft^2)

A_{pm} = pipe mean surface area (ft^2)

A_{po} = pipe outside surface area (ft^2)

COP = coefficient of performance

Cond = condenser

C_p = specific heat at constant pressure for air (Btu/(lb·°F))

DP = pressure drop (inches of water gage)

EER = energy efficiency ratio (Btu/Wh)

Evap = evaporator

h_i = inside-tube heat-transfer coefficient (Btu/(ft^2·h·°F))

h_l = heat-transfer coefficient for condensate (frost) layer (Btu/(ft^2·h·°F))

h_{pf} = heat-transfer coefficient for tube/fin contact (Btu/(ft^2·h·°F))

h_c = air-side heat transfer coefficient (Btu/(ft^2·h·°F))

K = material thermal conductivity (Btu/(ft·h·°F))

m_a = air mass flow rate (lb/h)

NTU = number of transfer units (non-dimensional)

P = pressure (psi)

P_{crit} = critical pressure (psia)

Q = capacity (Btu/h)

rmass = refrigerant mass flow rate (lb/h)

SEER = seasonal energy efficiency ratio (Btu/Wh)

s = entropy (Btu/(lb·°F))

T = temperature (°F)

T_{crit} = critical temperature (°F)

T_{sat} = saturation temperature (°F)

T_{sup} = superheat (°F)

TXV = thermostatic expansion valve

UA = heat-transfer conductance (Btu/(h·°F))

X_p = thickness of the tube wall (ft)

a = $i_{fgw}(\omega_a - \omega_w)/(C_{pa}(T_a - T_w))$

ε = heat-transfer effectiveness (fraction)

ϕ = fin efficiency (fraction)

ω_{ai} = humidity ratio of air at tube inlet $(lb_w/lb_{a,dry})$

ω_{ao} = humidity ratio of air at tube outlet $(lb_w/lb_{a,dry})$

ω_{fm} = humidity ratio of saturated air at mean temperature of condensate wetting the fin $(lb_w/lb_{a,dry})$

ω_ω = humidity ratio of saturated air at temperature of condensate wetting the tube $(lb_w/lb_{a,dry})$

Subscripts

a = air

o = outside

P = pipe

sat = saturated

vol = volumetric

w = water

1. SCOPE OF THE STUDY

In September 1999, the Working Fluids Subcommittee of the ARTI-21CR program identified R410A performance at high temperatures to be a very high research priority. Consequently, a research project was arranged covering broad research needs related to application of R410A in unitary equipment. The three distinct tasks were formulated and assigned to three research teams as follows:

Task 1. Refrigerant Property Measurements and Modeling

Principal Investigator: Dr. Mark O. McLinden

Physical and Chemical Properties Division

National Institute of Standard and Technology

Project Number: ARTI-21CR/605-50010-01-Pt. 1

This task encompassed selected property measurements for R125 and R410A. These new data and additional literature data would provide a basis for further improvement of REFPROP's robustness and predictions for R410A.

Task 2. Air Conditioner and System Study

Principal Investigator: Dr. Piotr. A. Domanski

Building Environment Division

National Institute of Standards and Technology

Project Number: ARTI-21CR/605-50010-01-Pt. 2

This task consisted of experimental and modeling parts. The experimental part included laboratory tests of R22 and R410A units at a wide range of ambient temperatures. The specific interest in high ambient temperatures was due to the low critical temperature of R410A, which may results in transcritical operation of the R410A system on extremely hot summer days. The tests were to allow performance comparison of **R22** and R410A systems and to observe operation of the R410A system while working in the transcritical cycle regime.

The modeling **part** of Task **2** included (1) development of REFPROP6-based models for finned-tube evaporators and condensers, EVAPS and COND5, (2) implementation of these simulation models into a user-friendly EVAP-COND simulation package, and (3) simulations of R22, R410A, R134a, and R404A systems at typical and elevated ambient temperatures for performance comparison using a reactivated NIST heat pump simulation model.

Task 3. Modeling, Validation, and Analysis of Sub- and Transcritical Performance of R410A Under Extreme Air Conditioning Conditions

Principal Investigator: Dr. C. Keith Rice

Oak Ridge National Laboratory

Project Number: ARTI-21CR/605-50015-01

Task 3 stipulated detailed validating/calibrating of the DOE/ORNL Heat Pump Model against the test data obtained at NIST under Task 2.

This document covers the work carried out under Task 2. Task 1 and Task 3 are covered by separate reports.

2. IMPACT OF ELEVATED AMBIENT TEMPERATURES ON CAPACITY AND ENERGY INPUT TO A VAPOR COMPRESSION SYSTEM – LITERATURE REVIEW*

2.1 Theoretical Background

Operation of a system at elevated ambient temperatures inherently results in a lower coefficient of performance (COP). This conclusion comes directly from examining the Carnot cycle. The COP relation, $COP=T_{evap}/(T_{cond}-T_{evap})$ indicates that the COP decreases when the condenser temperature increases at a constant evaporation temperature. This theoretical indication derived from the reversible cycle **is** valid for all refrigerants. For refrigerants operating in the vapor compression cycle, the COP degradation is greater than that for the Carnot cycle and varies among fluids.

The two most influential fundamental thermodynamic properties affecting refrigerant performance in the vapor compression cycle are refrigerant's critical temperature and molar heat capacity. (e.g., McLinden, 1987, Domanski, 1999). For a given application, a fluid with a lower critical temperature will tend to have a higher volumetric capacity (Q_{vol}) and a lower coefficient of performance (COP). The difference between COPs is related to different levels of irreversibility because of the superheated vapor horn and the throttling process, **as** shown conceptually in Figure 2.1. The levels of irreversibility vary with operating temperatures because the slopes of the saturated liquid and vapor lines change, particularly when approaching the critical point.

* Authored by S.Y. Motta and P.A. **Domanski,** this section was submitted to ARTI **as** a letter report in August 2000.

Refrigerants with a low critical temperature have a high pressure, a low drop of saturation temperature for a given pressure drop, and a low condenser-to-evaporator pressure ratio. These properties offer some advantages, which can be exploited in a real system for the betterment of its performance. Some researchers reported that a low pressure ratio promotes an improved compressor isentropic efficiency (e.g., Rieberer and Halozan, 1998). The low drop of refrigerant saturation temperature for a given pressure drop ($dT/dP|_{sat}$) allows designing heat exchangers with a high refrigerant mass flux, which promotes an improved refrigerant-side heat transfer coefficient.

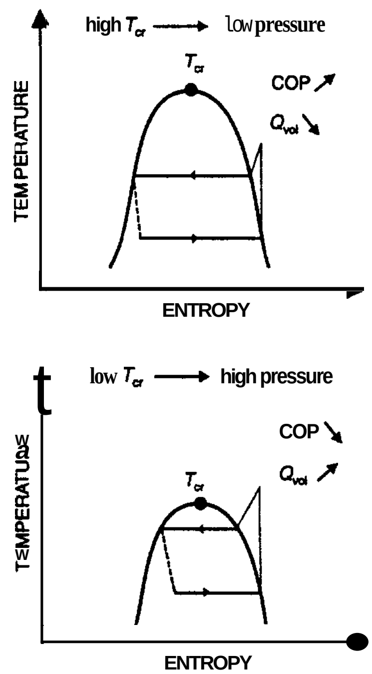


Figure **2.1** Impact of critical temperature on system performance

The condenser temperature increases at elevated ambient temperatures, which causes changes in refrigerant transport properties. These changes do not override the thermodynamic consideration, but they should be noted to foster complete understanding of the phenomena involved. The changes of liquid viscosity, conductivity, and heat capacity are smooth and favorable while approching the critical temperature (viscosity decreases, conductivity and heat capacity increase). In the supercritical region, density has a smooth transition above the critical point, but specific heat has a pronounced peak, as Figure **2.2** shows for R410A (Bullock, 1999). This trend in the neighborhood of the critical point is typical for all fluids as has been recently presented for carbon dioxide in several studies (e.g., Olson, 1999 who showed that conductivity and viscosity have a smooth transition as well). Because of the abrupt change in specific heat (Figure 2.2), the heat transfer coefficient at constant pressure (Figure 2.3) has a peak while approaching the critical temperature.

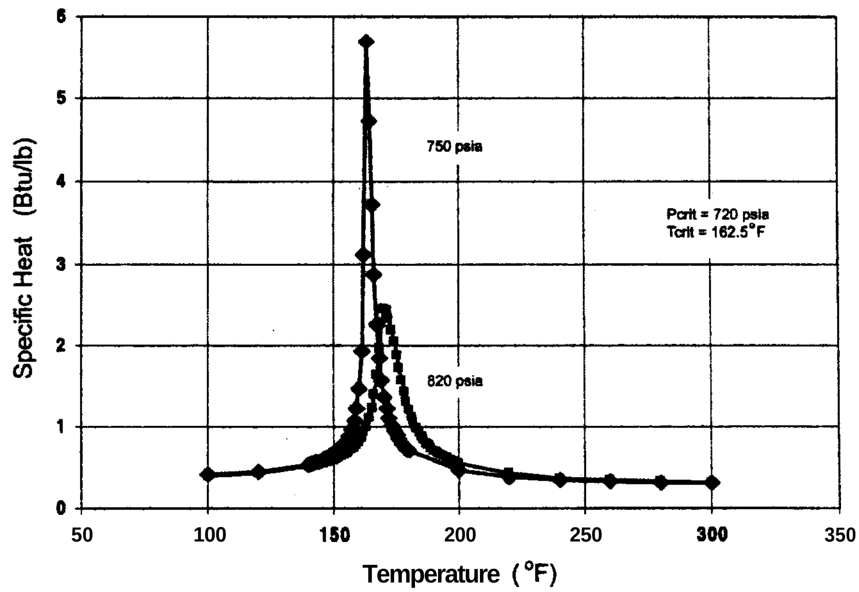


Figure 2.2 Refrigerant specific heat versus temperature and pressure: R410A (Bullock,1999)

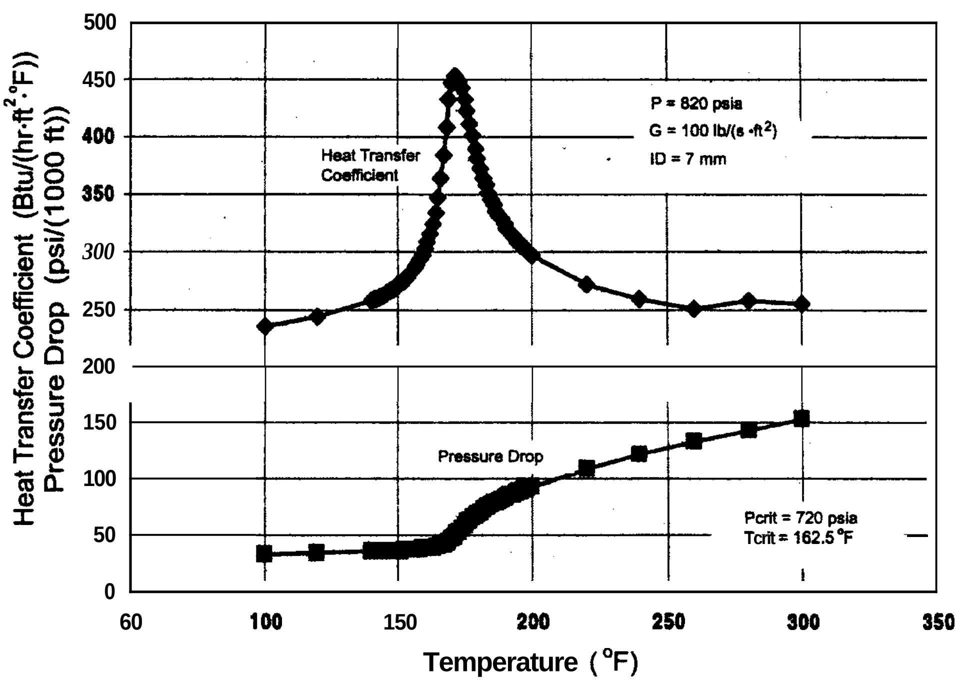


Figure 2.3 Refrigerant pressure drop and convection heat-transfer coefficient for supercritical flow of R410A (Bullock, 1999)

2.2 Literature Review

We were able to locate only a few publications concerned with air conditioner operation at elevated temperatures. They are reported here along with two seminar presentations made during the ASHRAE summer meeting in 1999. LeRoy et al. (1997) investigated capacity and power demands of R22 unitary systems under extreme operating conditions. The main goal of the study was to validate performance predictions of three public-domain heat pump simulation models. The authors used data of ten systems from tests at the 95.0 "F (35.0 "C) rating point and at higher outdoor temperatures. Three of these systems were tested at 115.0 "F (46.1 "C) and another three at 125.0 °F (51.7 "C) with the same indoor conditions of 80.0 "F (26.7 "C) dry-bulb and 67.0 "F (19.4 "C) wet-bulb temperature. The reported decrease in capacity at 115.0 "F (46.1 °C) was in the 14 % to 19 % range while the decrease in the energy efficiency ratio (EER) was in the 24 % to 41 % range. At 120.0 "F (48.9 "C), the capacity and EER decreases were within the 11 % to 20 % range and 34 % to 39 % range, respectively. These data indicate that performance degradation at high ambient temperature varies significantly from one system to another.

Chin and Spatz (1999) explored some of the advantages and disadvantages of R410A use in air conditioning systems. They used compressor performance data and a heat pump simulation model to compare R22 and R410A. In this study, they also performed heat exchanger optimization to exploit the favorable thermophysical properties of R410A. The authors reviewed experimental heat transfer and pressure drop data for R22 and R410A in evaporation and condensation processes. Figure 2.4 helps to explain the authors' findings. As a reference, they used the R22 pressure drop and saturation temperature drop at a mass flux of **147158** lb/(h·ft^2) (**200** kg/(s·m^2)). For these conditions, **R410A** requires a **mass** flux of **206022** lb/(h·ft^2) (**280** kg/(s·m^2)) to match the **R22** pressure drop **and** a mass flux of **250170** lb/(h·ft^2) (**340** kg/(s.m^2)) to match the **R22** drop in saturation temperature. If the **R410A** mass flux is selected to match the **R22** drop in saturation temperature, **R410A** will have a **55 %** higher heat transfer coefficient than **R22.**

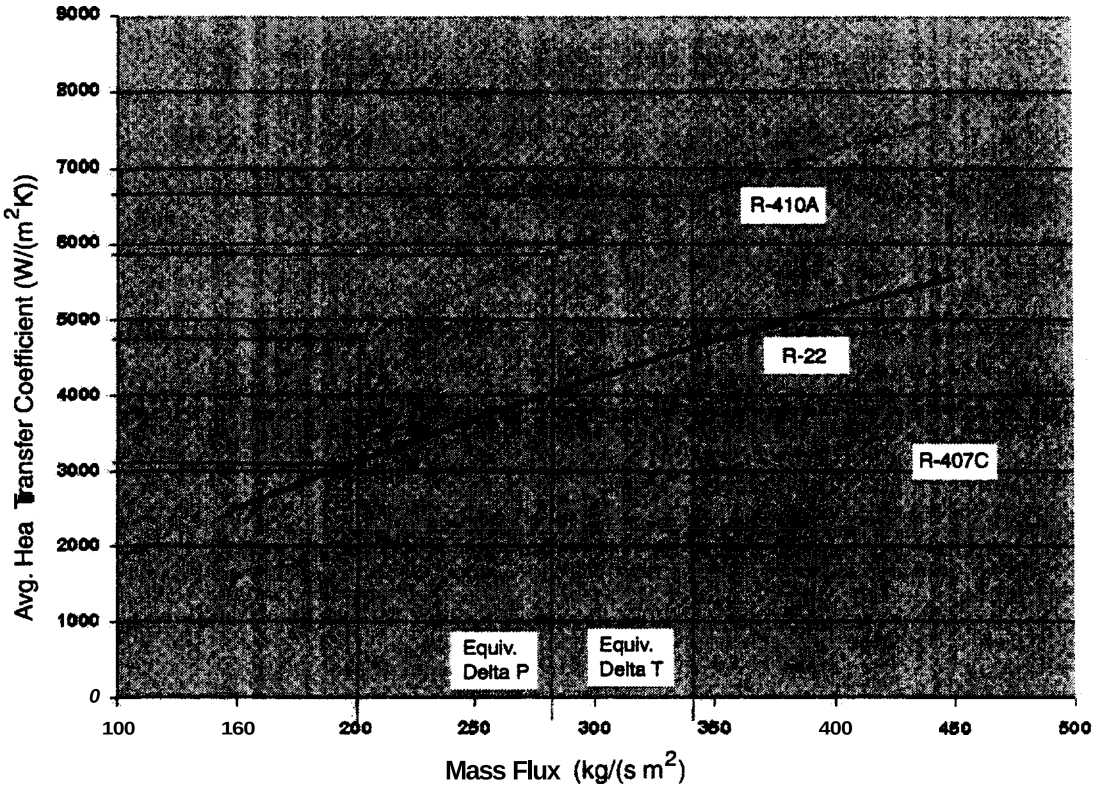


Figure **2.4** Heat transfer – evaporation (Spatz, **2000)**

Table **2.1** Capacity and **COP** of **R22** and **R410A** systems **as** function of outdoor temperature (Chin and Spatz, **1999)**

Ambient Air Temperature		28 °C (82°F)	35 °C (95°F)	46 °C (115°F)	52 "C (125 °F)	57 °C (135 °F)
Capacity (kW)	R22	12.84	11.98	10.63	9.95	9.32
	R410A	13.01	11.92	10.20	9.32	8.50
	Rel* (%)	1.3%	-0.5%	-4.0%	-6.3%	-8.8%
COP	R22	3.79	3.11	2.26	1.92	1.64
	R410A	3.99	3.19	2.19	1.79	1.47
	Rel* (%)	5.3%	2.4%	-3.4%	-6.9%	-10.7%

After the evaporator and condenser were optimized, Chin and Spatz performed for **R22** and **R410A** simulations system. Table **2.1** shows their capacity and **COP** results. The authors concluded that the superior performance of the **R410A** compressor and optimized heat exchangers compensated for the lower thermodynamic efficiency of **R410A** relative to **R22** at low and moderate condensing temperatures. However, the **R410A** optimized-system experienced a loss in COP relative to the **R22** system at condensing temperatures exceeding **116.6** "F **(47.0** "C).

Meurer et al. **(1999)** compared the performances of **R22** and **R410A** working at elevated condensing temperatures up to **140.0 °F** (60.0 "C) in a breadboard apparatus. The components of the system were an open reciprocating compressor, **a** water-cooled condenser, a methanol-heated evaporator, a thermostatic expansion valve, and a liquid-line accumulator. The authors reported the **R410A** compressor having higher isentropic

(+14 %) and volumetric (+22 %) efficiencies than **R22.** For a typical evaporation temperature of **48.2 °F** (9.0 "**C),** the COP of **R410A** was higher by **16 %** at a condensing temperature of **80.6 °F (27.0 "C),** but it was lower by 1 % at a **134.6 °F (57.0 "C)** condensing temperature. The authors stated that a lower compressor speed accounted for part of the benefits measured with **R410A,** but the use of equal rotational speed would negatively affect the **R410A** compressor and system performance.

Wells et al. **(1999)** compared the performance of **R410A** and **R22** in split and window-type air conditioners. Their study included theoretical simulations, laboratory testing of split systems, laboratory testing of window units with several hardware modifications, and simulations using the ORNL heat pump model. Figures **2.5** and **2.6** show the capacity and **EER** trends obtained from the **R22** and **R410A** split system tests referenced to the respective values at a **95.0 °F (35.0 °C)** ambient temperature. **At** an ambient temperature of **125.0** "F **(51.7 "C),** the capacity and **EER** ratios of **R410A** fell **12 %** below that of **R22.** Similar results (within the data scatter) were obtained for the window units. Increased subcooling benefited performance at high ambient temperatures. **The** study also concluded that using a TXV versus a short tube restrictor or capillary tube results in less performance loss.

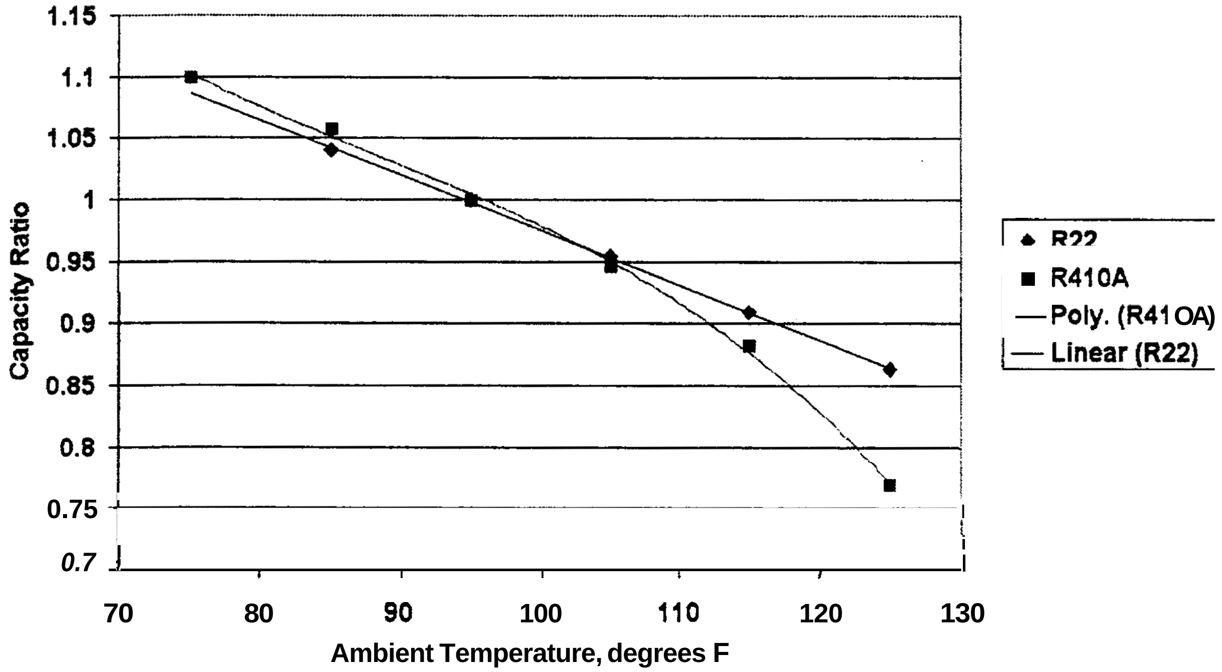


Figure 2.5 Comparison of capacity **loss** versus ambient temperature, split system A/C, 12-13 **SEER** (Wells et al., 1999)

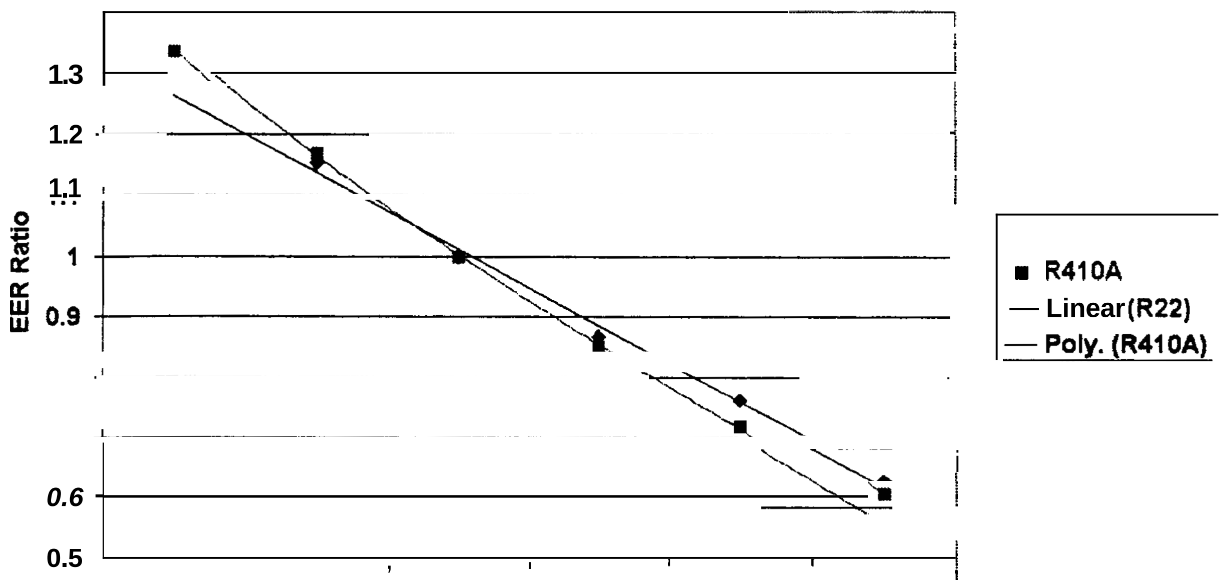


Figure 2.6 Comparison **of** EER loss versus ambient temperature, split system A/C, 12-13 SEER (Wells et al., 1999)

Bullock (1999) investigated the performance of **HVAC** systems working with two low-critical temperature refrigerants: R404A and R410A. The study included theoretical analysis of **the** refrigerant properties, simulations of the basic thermodynamic cycle, and simulations of three split systems: two using R410A and one using R404A. The main difference between the systems studied was the condenser and blower size. In Bullock's A/C simulation model, the compressor, expansion device, and condenser/gas cooler models were modified to accommodate transcritical system operation.

Figure **2.7** presents simulation results for one of the systems studied by Bullock (1999). The vertical arrow indicates the outdoor temperature at which the condenser pressure exceeded that of the critical point. The simulations show that the capacity degradation and compressor power increase become more significant with an increase of outdoor temperature when the condenser pressure is above the critical point. Based on simulation results from the three systems, Bullock offered the following key conclusions: a typical unitary system will cross over to transcritical operation at about 135.0 °F to 140.0 °F (**57.2 °C** to **60.0** °C). At the ambient temperature when the critical point is reached, the cooling capacity will be about 60 % to **70** % of the capacity at the 95.0 °F (35.0 °C) rating point, and the compressor power will be about 110 % to 160 % of the power at the 95.0 °F (35.0 °C) rating point (depends greatly on the compressor type). The system performance at high ambient temperatures can be improved by providing an oversized condensing unit.

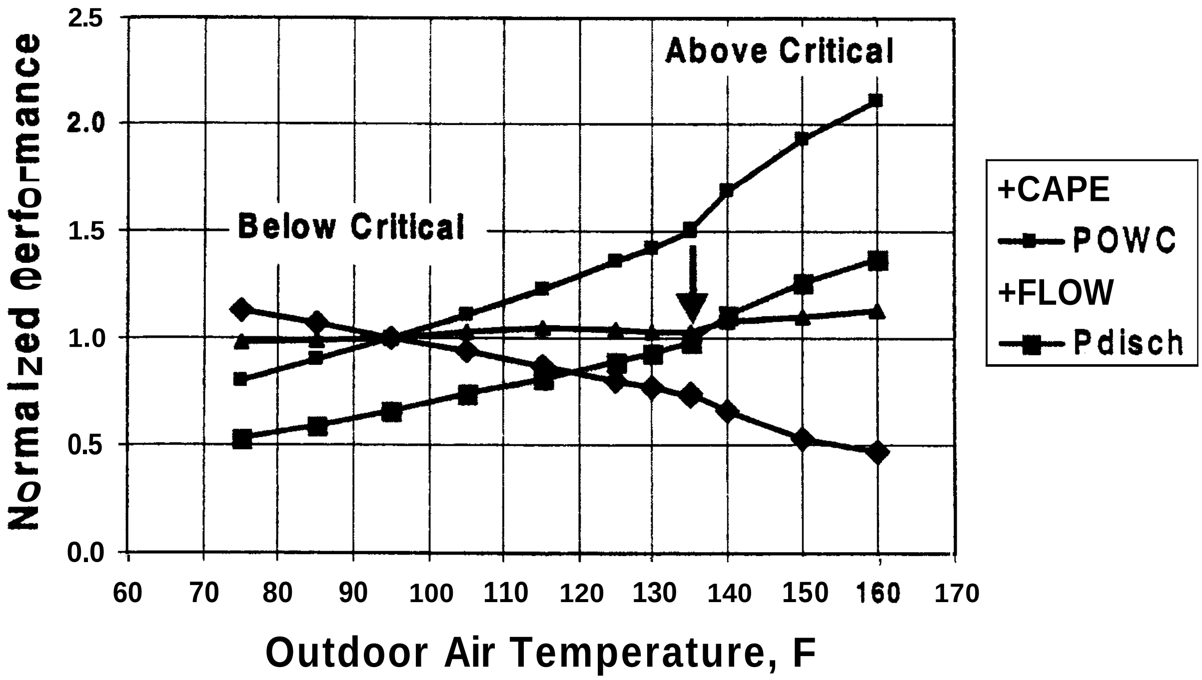


Figure 2.7 Performance map for R410A unit with a high performance NTU(0.9) and a low condenser cfm/ton, (640). **CAPE** = capacity of evaporator; POWC = power of compressor; **FLOW** = refrigerant flow rate; Pdisch = compressor discharge pressure (all normalized to their values at 95.0 °F (35.0 °C), except for the compressor discharge pressure, which is related to the critical pressure); **(Bullock,** 1999)

Yana Motta and Domanski (2000) performed a simulation study to evaluate capacity and COP of an air conditioner working with R22, R134a, R290, R410A, and R407C. Figures 2.8 and 2.9 present two-phase domes of the studied refrigerants with the horizontal axes using non-dimensional entropy, s^*, and enthalpy, h^*, respectively (where $s^*=(s-s^0_l)/(s^0_v-s^0_l)$, $h^*=(h-h^0_l)/(h^0_v-h^0_l)$, s, h = entropy and enthalpy, s^0_v, h^0_v = entropy and enthalpy of saturated vapor at 0 °C (32 °F), and s^0_l, h^0_l = entropy and enthalpy of saturated liquid at 0 °C (32 °F)). These figures are suitable for qualitative analyses of the impact of the shape of the two-phase dome on the COP because the width of the two-phase dome is

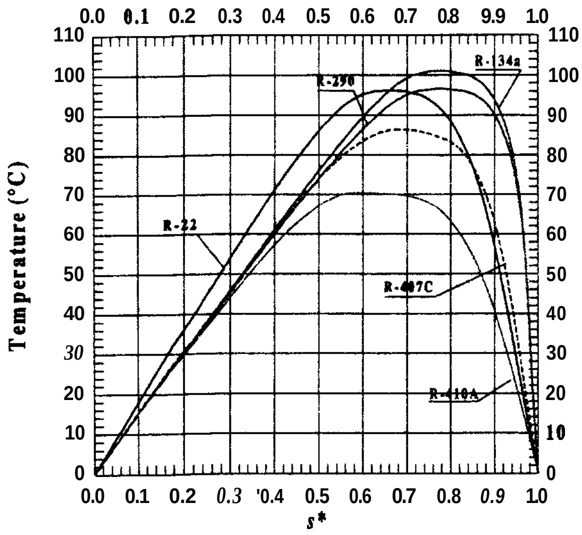


Figure **2.8** Temperature-dimensionlessentropy diagram (Yana Motta and Domanski, **2000).**

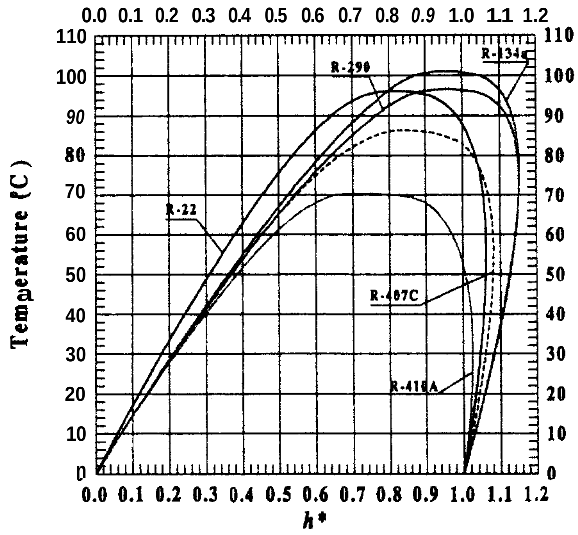


Figure **2.9** Temperature-dimensionless enthalpy diagram (Yana Motta and Domanski, **2000).**

normalized. If we envision vapor-compression cycles with their corresponding Carnot cycles drawn for each refrigerant with the same condensing and evaporating temperatures, we can conclude that the superheated-vapor horn irreversibilities (Figure **2.8)** and throttling-induced capacity losses (Figure **2.9)** will be greater for **R410A** than for **R22** due to **R410A's** smaller two-phase dome.

Yana Motta and Domanski simulated performance of different refrigerants using the **UA** version of NIST's semi-theoretical vapor-compression model CYCLE-11 (Domanski and McLinden, **1992).** All system components were the same for the five fluids, except the compressor for which the swept volume was adjusted to obtain the same capacity at the **95.0 °F (35.0 °C)** rating point for each fluid. **A** reference scheme was used to account for different transport properties and their impact on heat transfer coefficient and pressure drop for the different refrigerants.

Figure 2.10 shows changes of COP for each refrigerant for different outdoor temperatures. The COP values are normalized by the COP at 95 °F (35 "C) for each fluid. R410A **has** the highest degradation in COP and R134a has the lowest. The lines representing performance of R410A (the lowest-critical-temperature fluid) and R134a (the highest-critical-temperature fluid) bracket the performance of the remaining refrigerants. The change of COP for R22, R290, and R407C is **very** similar, because their critical temperatures are within 18.0 "F (10.0 °C) of each other.

Figure 2.11 presents the COP of the four alternatives normalized by the COP of the R22 system. R134a, the fluid with the highest critical temperature, improves its performance in relation to R22, and it hasa higher COP **than** R22 at outdoor temperatures greater than 95 °F (35 "C). On the other hand, the COP of R410A drops dramatically at increasing outdoor temperature. Regarding the fluids with critical temperatures similar to R22 (R407C and R290), the small COP differences are caused by the different shapes of the two-phase domes of these fluids rather than their different critical temperatures.

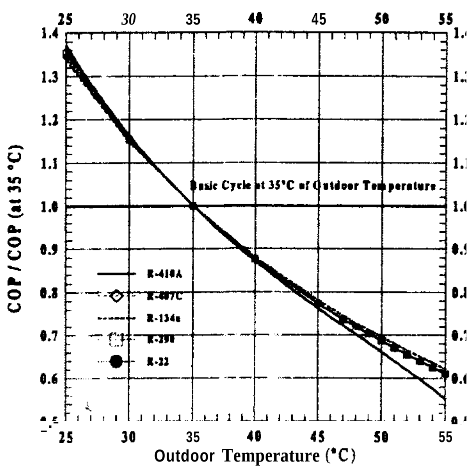


Figure 2.10 COP referenced to COP at 95 °F (35 °C) (Yana Motta and Domanski, 2000).

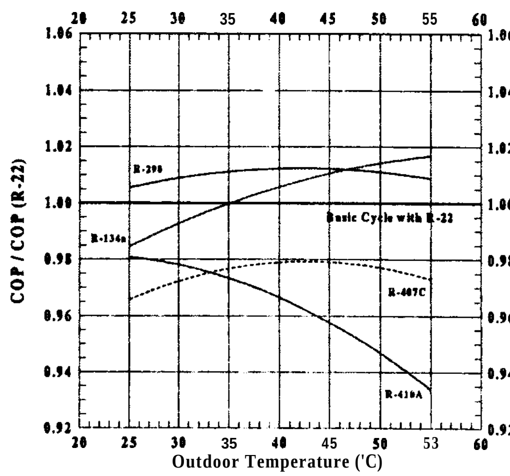


Figure 2.11 COP referenced to COP of R22 system (Yana Motta and Domanski, 2000).

Yana Motta and Domanski also evaluated the impact of using a liquid-line/suction-line heat exchanger (llsl-hx). As Figure 2.12 shows, the use of llsl-hx provided COP improvement for all fluids. Refrigerants having high molar capacity benefited more **with** the llsl-hx application. The benefit **of** llsl-hx for R410A increased slightly at high ambient temperatures due to a change in the slope of the saturated liquid line while approaching the critical point; however, the overall impact of approaching the critical point was not significant. At an outdoor temperature of 131.0 °F (55.0 °C), the COP increase due to the llsl-hx **was** 1.9% higher for R410A than that for R22.

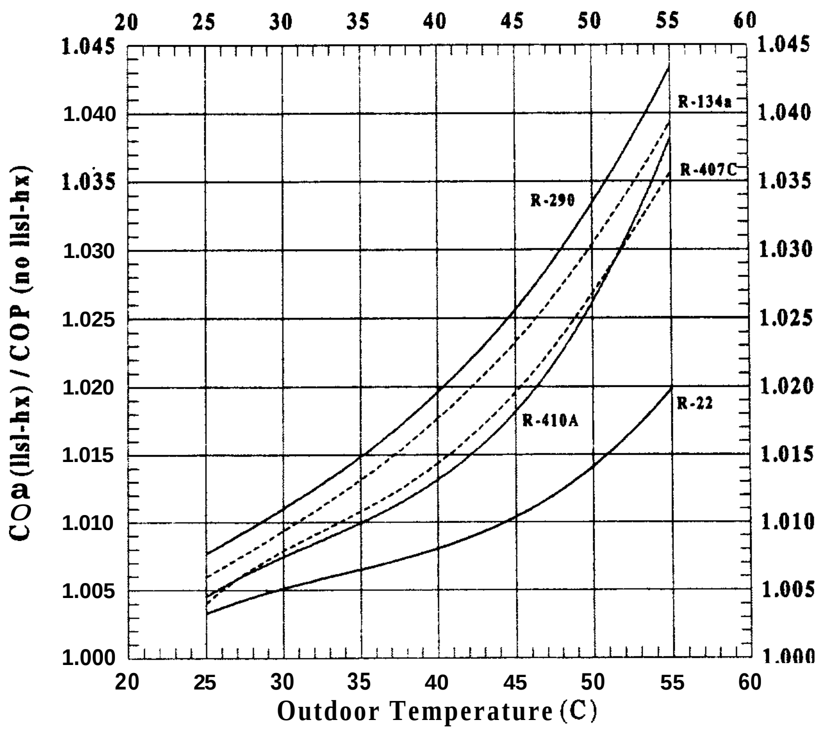


Figure 2.12 COP for llsl-hx cycle referenced to COP for basic cycle (Yana Motta and Domanski, 2000).

2.3 Concluding Remarks

1. Operation of a vapor compression system at elevated ambient temperatures inherently results in a lower COP. For refrigerants operating in the vapor compression cycle, the COP degradation is greater than that for the Carnot cycle and varies between fluids. The refrigerant-related factors that most influence the degradation are the critical temperature and the shape of the two-phase dome.

2. Degradation of capacity and COP at high outdoor temperatures can vary significantly between systems. The system design (size of the condenser, refrigerant charge, refrigerant expansion device) influences performance degradation.

3. All experimental and simulation studies reported a loss of performance for R410A systems at elevated ambient temperatures by approximately 10 % as compared to R22.

4. Simulation results indicate that the use of llsl-hx provides slightly better improvement of COP for R410A than for R22. At an outdoor temperature of 131.0 °F (55.0 °C), the COP increase for R410A was 1.9 % higher than for R22.

5. The thermodynamic loop of a typical unitary R410A A/C will cross above the critical point at an outdoor temperature of approximately 135.0 °F (57.2 °C).

CHAPTER 3. LABORATORY EXPERIMENT

3.1 Units Selected for Testing

The systems consisted of 3-ton nominal cooling capacity units with scroll compressors, finned tube condensers, and finned tube evaporators. Manufacturer data listed the **R22** system with a **SEER** of **12.5** and the R410A system with a **SEER** of **13.** Both the **R22** system and the R410A system had identical evaporator and condenser coils. Only the thermostatic expansion valve and liquid line filter differed between the two system's piping arrangements. Figure **3.1** below shows the circuiting of the condenser finned tube coil. The condenser is **28** in **x 80.5** in **(71.1** cm x 204..5 cm) finned length, **22** fins/in (9 fins/cm).

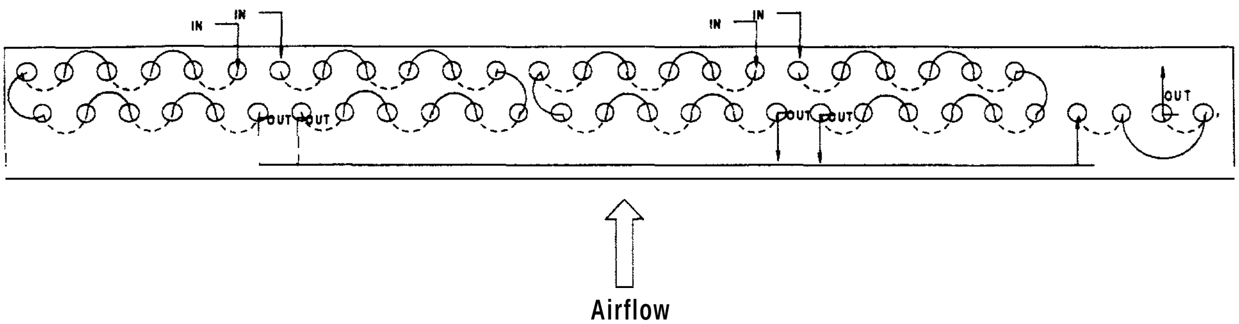


Figure **3.1** Condenser for **R22** and **R410A** systems

The evaporator for both systems was a vertical slab coil designed for installation with airflow from the left or right. Figure **3.2** shows the circuiting of the evaporator used by both systems. For all of the tests, the airflow rate through the evaporator was set at **1200.0** scfm **(34.0** m^3/min). The evaporator is **22.0** in **x 26.0** in **(55.9** cm **x 66.0** cm) finned length with **12** fins/in. The evaporator and condenser have a fin thickness of 0.0045 in **(0.1143 mm)**.

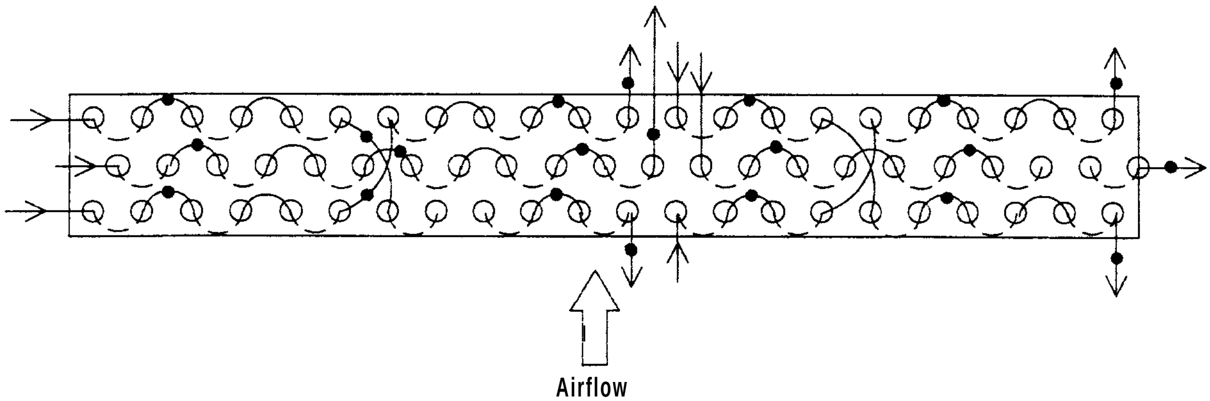


Figure **3.2** Evaporator for **R22** and **R410A** systems

3.2 Experimental Set-Up

Figure **3.3** shows the arrangement of the system in the environmental chambers. The airflow chamber contained a **7.0** inch **(17.8** cm) diameter **ASME** nozzle and **was** constructed according to **ANSI/ASHRAE 51-1985** and **ANSI/ASHRAE** Standard **37-1988.** The air conditioning systems consisted of a condensing unit in the outdoor chamber (Figure **3.4)** and a finned tube coil evaporator in the indoor test section (Figure 3.5). Air was pulled through the evaporator by a centrifugal fan at the outlet of the nozzle chamber ductwork. Dew-point temperature was measured at the inlet of the evaporator ductwork and in the ductwork after the evaporator once the air passed through several mixers. Twenty-five node thermocouple grids and thermopiles measured the air temperatures and temperature change, respectively. The thermocouple grids were used to ensure that the ***air*** was well mixed before and after the evaporator. Barometric pressure, evaporator air pressure drop, nozzle pressure drop, and nozzle temperature were used along with the dew-point measurements to establish the thermodynamic state of the air. The air enthalpy method was used at the primary measurement of air-side capacity. The refrigerant enthalpy method was used as the secondary measurement of capacity. These two measurements always agreed within 3.5 %.

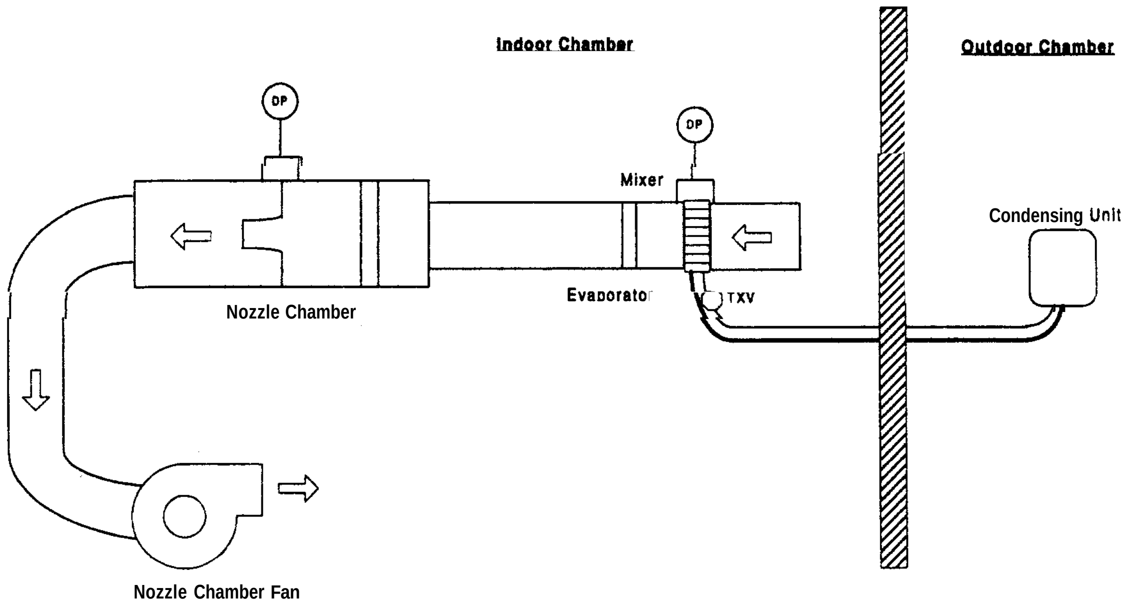


Figure 3.3 Environmental chamber test schematic

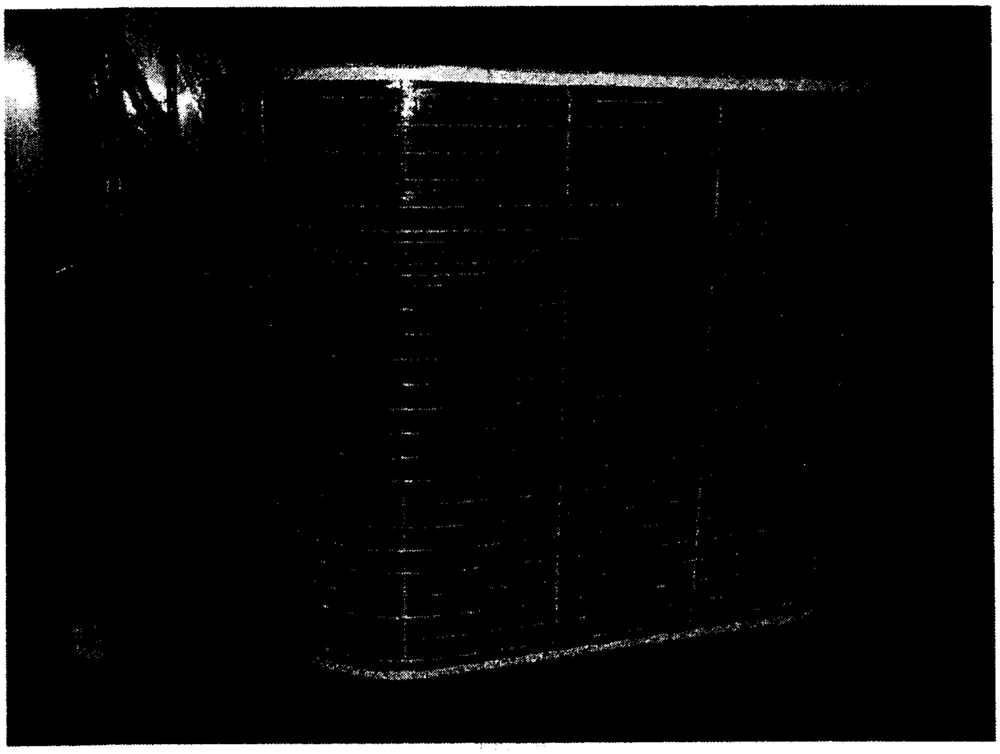


Figure 3.4 High efficiency condensing unit

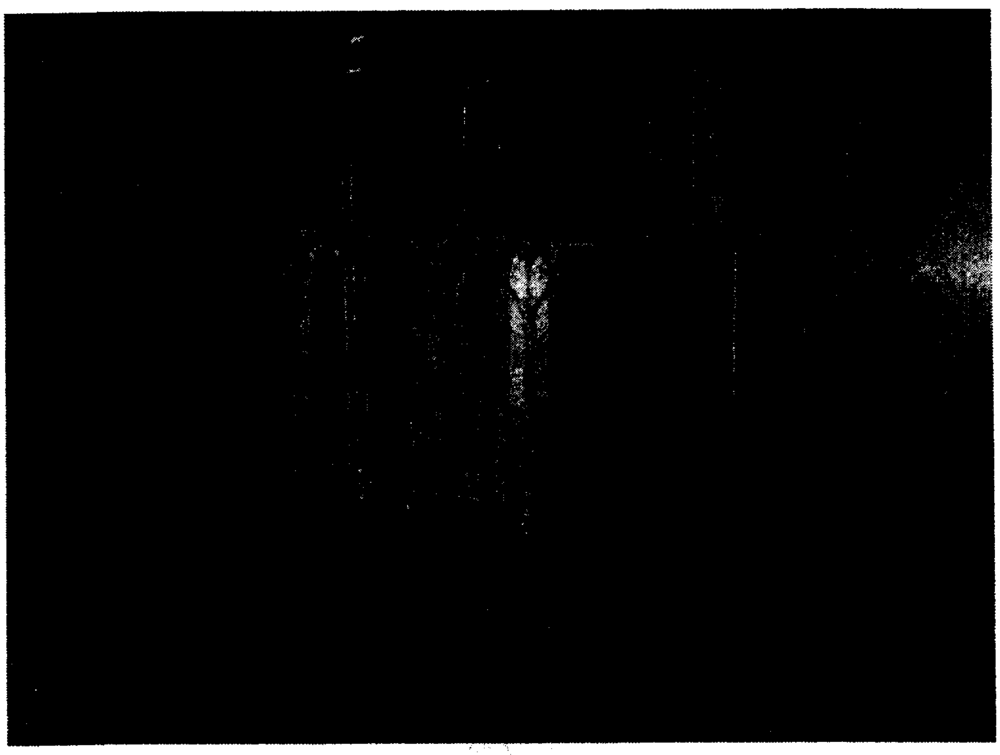


Figure 3.5 Indoor test section housing evaporator

3.3 Experimental Procedure and Test Conditions

The two units were tested at the same indoor conditions of 80.0 °F (26.7 °C) dry-bulb and 67.0 °F (19.4 °C) wet-bulb temperature according to ASHRAE Standard 37-1988. The outdoor conditions varied according to the table below.

Location	Cooling	
	Setpoint (°F)	Tolerance (°F)
Indoor Dry-bulb Temperature	**80.0**	±0.5
Indoor Dew-point Temperature	**60.4**	±0.5
Outdoor Chamber Temperature	82.0[1] 95.0[2] 115.0[2] 125.0[1] 130.0[2] 135.0[3] 140.0[4] 150.0[4] 152.0[4] 155.0[4]	±0.5
Evaporator Airflow, scfm	**1200**	

1) R22 and R410A Compressor #1
2) R22 and both R410A Compressors
3) Only R22
4) Only R410A Compressor #2

Indoor and outdoor conditions remained stable for one hour before data were taken. A steady flow of condensate from the evaporator was present for all tests. Before any testing began, the system charge was set according to manufacturer's recommendations. The charge for both units was set by measuring the difference between the liquid line temperature exiting the condensing unit and the outdoor air temperature (95.0 °F (35.0 °C)). This temperature difference was decreased by adding refrigerant or increased by removing refrigerant. For the R22 system this temperature difference was set at 5.0 °F (2.8 °C). For the R410A system this temperature difference was set at 6.0 °F (3.3 "C).

Experimental uncertainty was calculated using a propagation of uncertainty technique considering uncertainty in all the parameters associated with the capacity and EER (Payne and Domanski, 2001). Appendix D summarizes the propagation of errors approach that was used to determine the uncertainty in capacity and EER. The 95 % (two sigma) uncertainty in capacity and EER varied from 2.9 % to 3.5% and 3.5 % to 5.4 %, respectively.

3.4 Experimental Results

3.4.1 Test Results for the R22 System

The R22 system performed without any difficulty over the full range of outdoor temperatures. The sensible heat ratio remained at 0.8 ± 0.05 for all tests. Figure 3.6 shows the cooling capacity as a function of the outdoor temperature. Capacity ranged from 40201 Btu/h (11781 W) to 29711 Btu/h (8707 W) over the range of outdoor temperatures. This was a decrease of 26.1 % from the high value at 82.0 "F (27.8 "C) to the low value at 135.0 "F (57.2 "C).

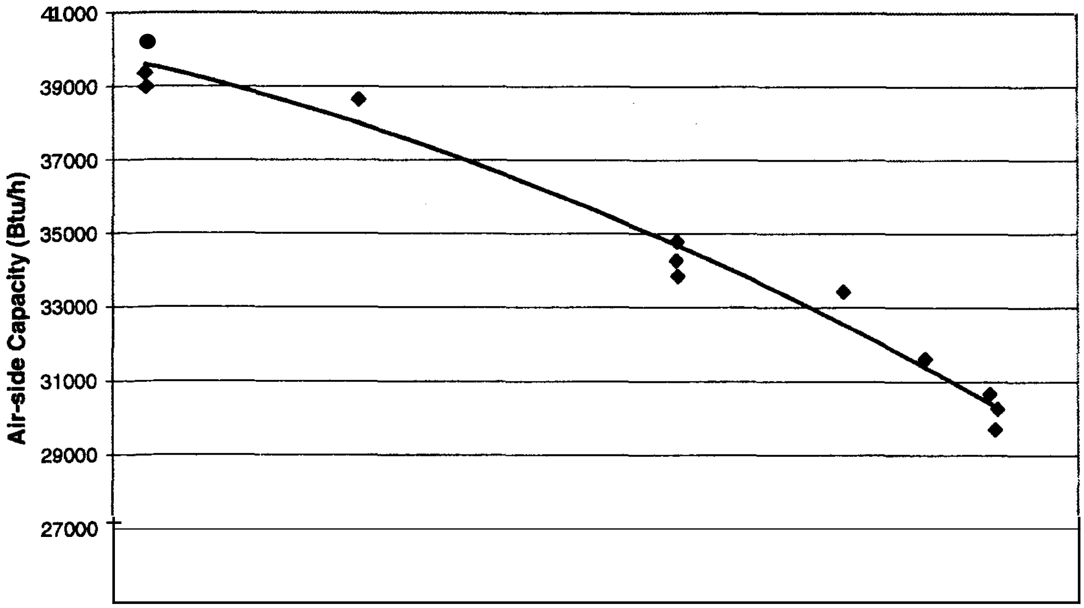


Figure **3.6 R22** cooling capacity as a function of outdoor temperature

The compressor performance was characterized by power measurements and refrigerant conditions at its inlet and exit. Figure **3.7** shows the reduced discharge pressure and discharge superheat for the tests shown in Figure **3.6.** Reduced pressure ranged from 0.30 to **0.56** with the discharge superheat ranging from 44.0 **°F** to **83.0 °F (24.4 °C** to **46.1 °C)** (P_{crit} = **723.7** psia **(4989.7** kPa), T_{crit}= **205.06 °F (96.14** "C)). The discharge pressure and discharge superheat increased by **84.4%** and **83.3** %, respectively, from their values at **82.0** "F **(27.8 °C)**.

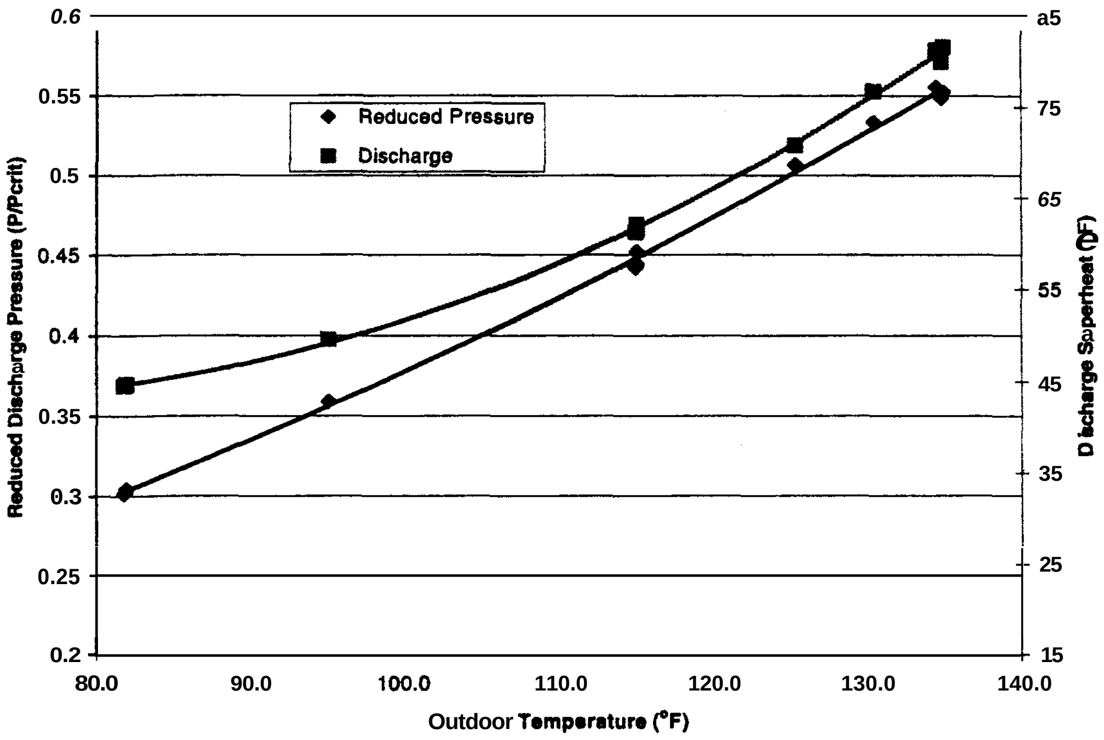


Figure 3.7 R22 reduced discharge pressure and discharge superheat as function of outdoor temperature

System power and refrigerant mass flowrate are shown in Figure 3.8. Power increased from 2080 W to 4140 W as the mass flowrate decreased from 8.93 lb/min (4.05 kg/min) to 8.54 lb/min (3.87 kg/min). This produced an increase of 87.9 % in system power with a 4.7 % decrease in refrigerant mass flowrate.

R22 cooling EER decreased as the outdoor temperature increased (Figure 3.9). As the outdoor temperature increased from 82 "F to 135 °F (27.8 °C to 57.2 °C), the EER (COP) decreased by 60.3 % as it dropped from 18.3 Btu/Wh (5.36) to 7.3 Btu/Wh (2.14).

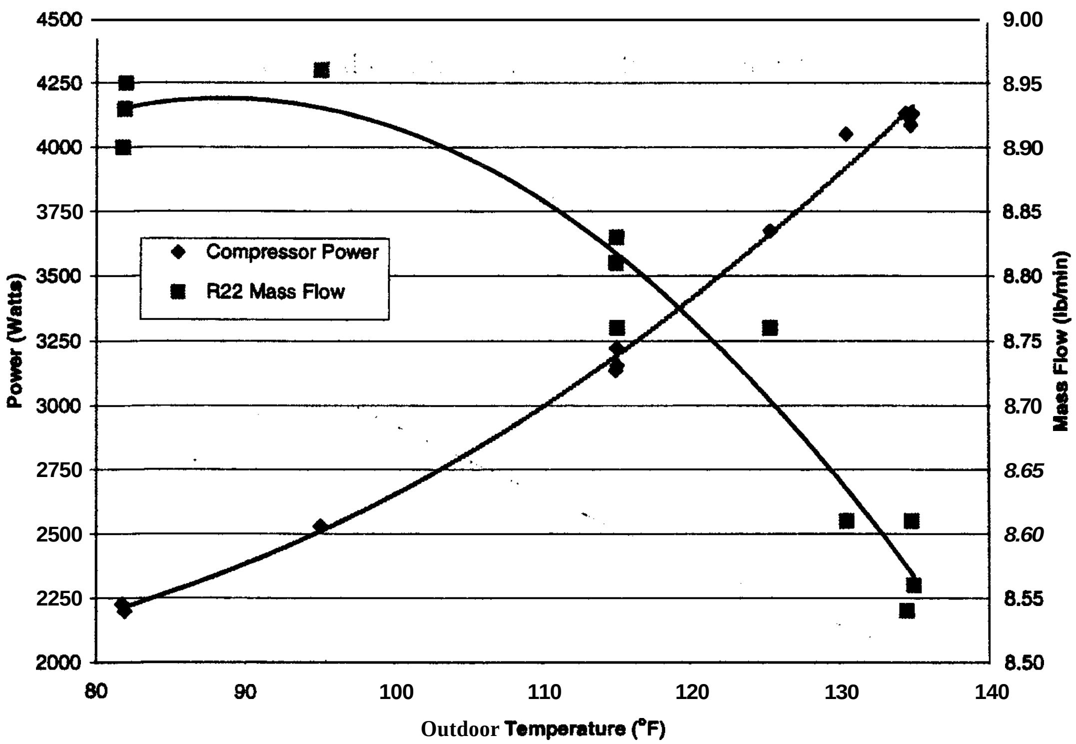
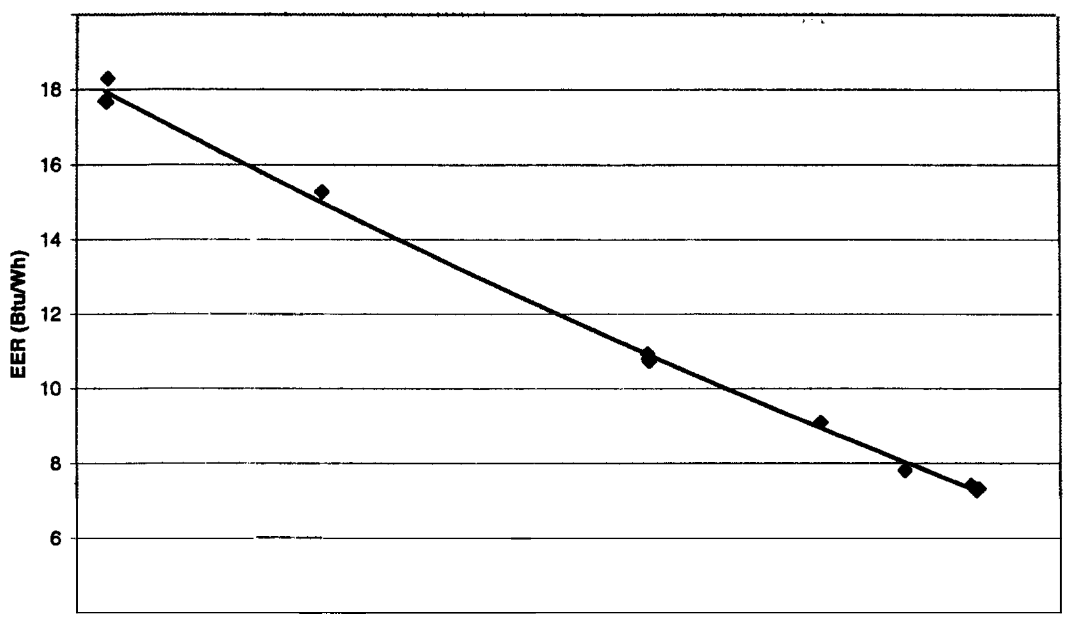


Figure 3.8 System power and R22 mass flow as a function of outdoor temperature



3.4.2 Test Results for the R410A System

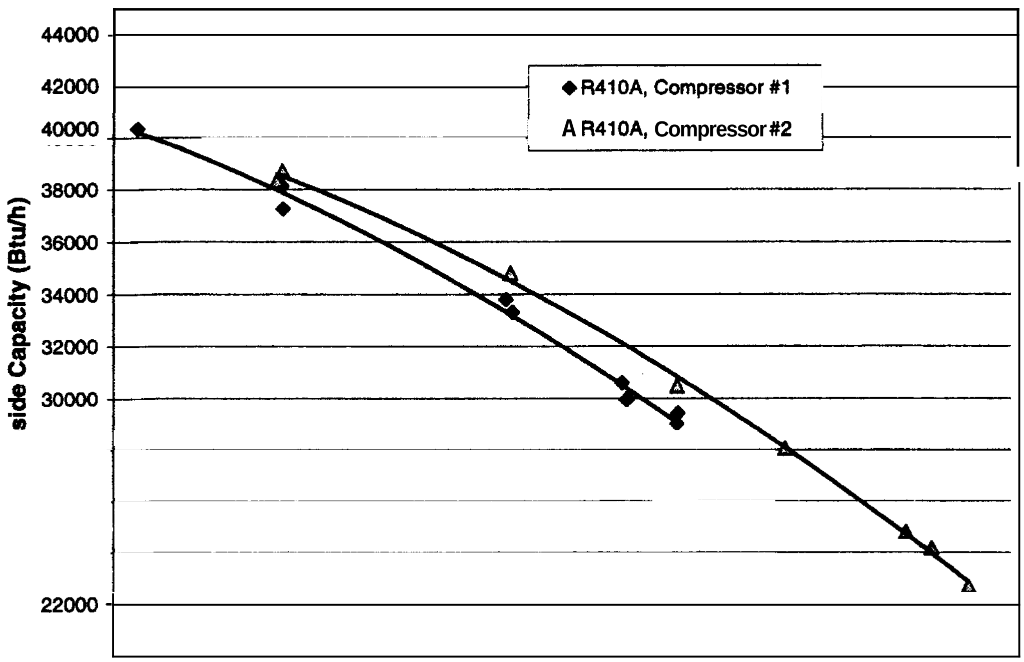
We performed R410A system tests using two compressors designated here as compressor #1 and compressor #2. Compressor #1 was the original compressor supplied with the system. The internal safeties for this compressor prevented the system from operating continuously at outdoor temperatures above 130.0 "F (54.4 "C). Another compressor that had all internal safeties removed was used to test the system at higher outdoor temperatures. Compressor #2 allowed testing to proceed up to 155.0 °F (68.3 "C). Compressor #2 had a more powerful electric motor than compressor #1.

The cooling capacity of the R410A system is show in Figure 3.10. Air-side capacity decreased in a nearly linear manner as the outdoor temperature was increased from 82.0 °F to 155.0 "F (27.8 "C to 68.3 "C). Over this temperature range, air-side capacity decreased from 40345 Btu/h (11824 W) to 22699 Btu/h (6652 W); a decrease of 43.7 %.

Figure 3.11 shows the compressor reduced discharge pressure and discharge superheat over the range of outdoor temperatures. The discharge pressure was above the critical pressure of 691.8 psia (4769.8 kPa) during three of the high ambient tests. Discharge superheat increased as the outdoor temperature increased and the refrigerant mass flowrate decreased. Compressor discharge superheat ranged from 22.0 "F (12.2 "C) at the lowest outdoor temperature to 97.0 "F (53.9 °C) at the highest outdoor temperature (for the tests above the critical temperature and pressure, superheat is calculated with respect to the critical temperature of 158.3 "F (70.2 "C), $T_{sup} = T - T_{crit}$).

R410A system power and refrigerant mass flowrate are shown in Figure **3.12.** The power increased from **2201** W to **6287** W **as** the mass flowrate decreased from **8.95** lb/min to **8.26** lb/min. This **was** an increase of **185 %** in system power with a **7.7 %** decrease in refrigerant mass flowrate.

R410A cooling EER decreased **as** the outdoor temperature increased (Figure **3.13). As the** outdoor temperature increased from **82.0 °F** to **155.0 °F (27.8 °C** to **68.3 °C),** the EER **(COP)** decreased **by 80.3 % as** it dropped from **18.3** Btu/Wh **(5.36)** to **3.6 Btu/Wh (1.06).**



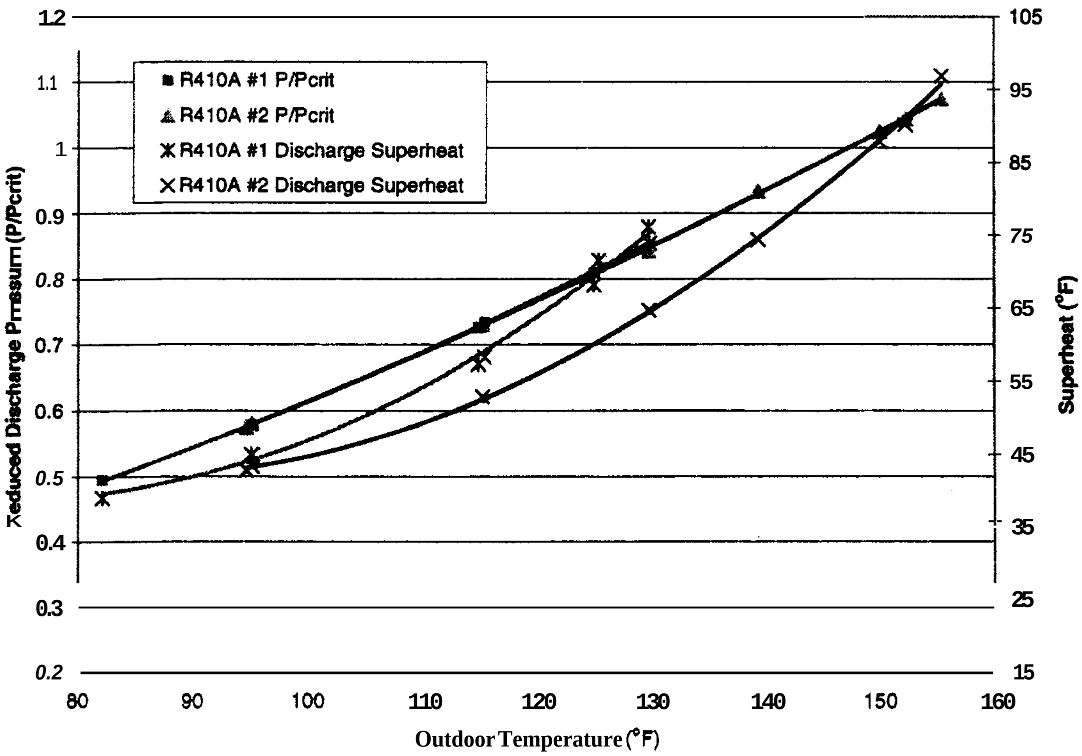


Figure 3.11 R410A reduced discharge pressure and discharge superheat

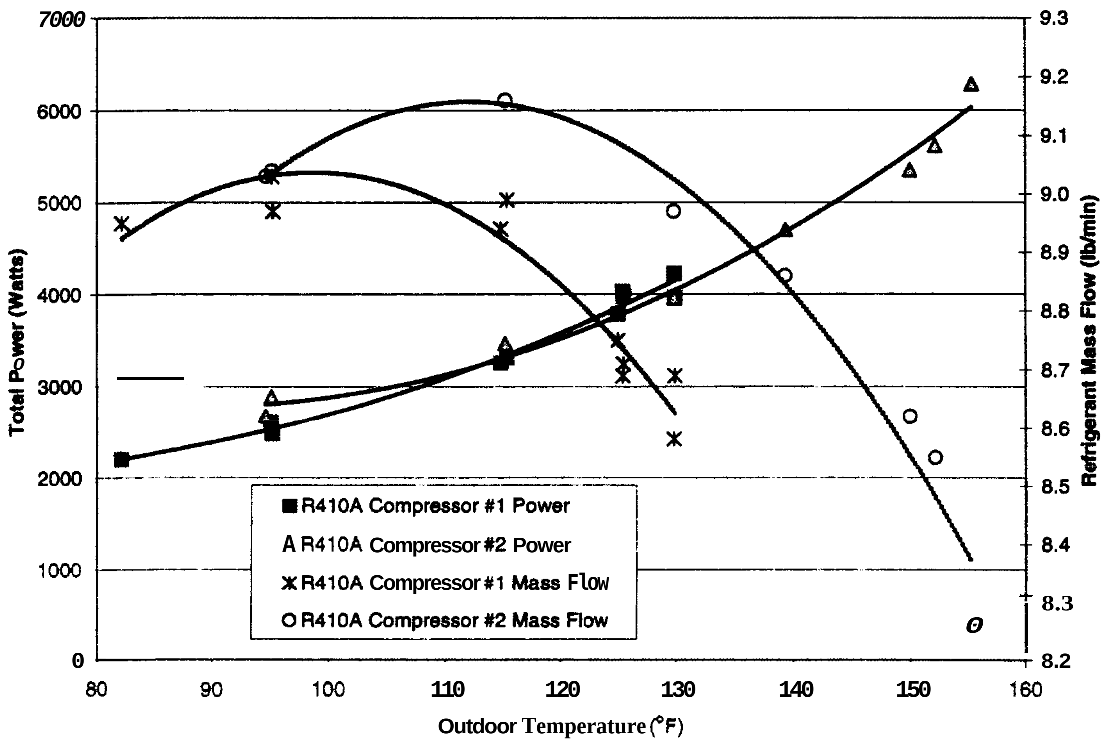


Figure 3.12 R410A system power and refrigerant mass flowrate

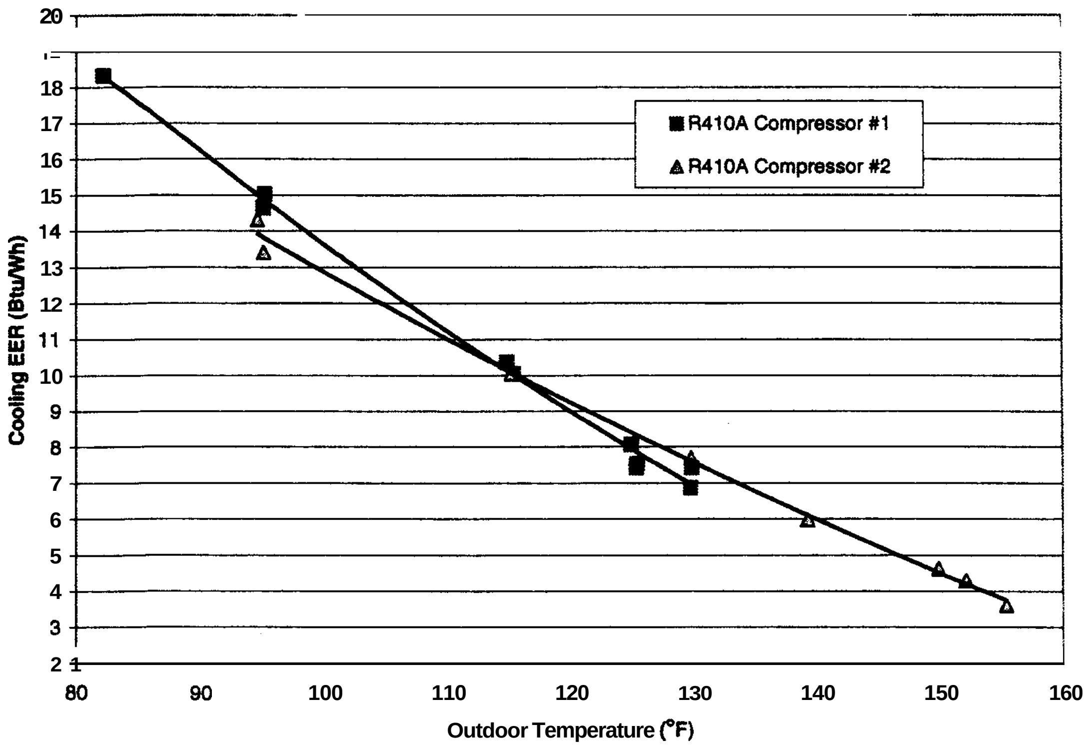


Figure 3.13 R410A cooling EER as a function of outdoor temperature

3.4.3 R410A Oil Sampling Test Results

POE oil (RL32S) was sampled from the R410A system at outdoor temperatures of 95.0 °F (35.0 °C) and 125.0 °F (51.7 °C). The sample cylinder was connected to the liquid line by a length of 1/8 inch (3.2 mm) copper tubing. The evacuated cylinder was submerged in an ice bath while a needle valve allowed a slow flow of liquid into the cylinder. Approximately 3 ounces (85 grams) of refrigerant and oil were sampled. The refrigerant was evacuated from the cylinder over a one-hour period. The cylinder was weighed before and after the sample was taken and after the refrigerant had been removed. The mass fraction of oil was defined as the mass of oil in the sample divided by the mass of refrigerant. The masses were recorded with an uncertainty at the 95 %

level of ± 0.000071 ounces (± 0.002 grams). Table 3.2 lists the temperatures and sample masses for the R410A tests on compressor #1.

Table 3.2 R410A oil sample results

Outdoor Temperature (°F)	Mass of Oil (gram)	Mass of Refrigerant (gram)	(Mass of Oil) / (Mass of Refrigerant)
95	0.456	86.438	0.53 %
95	0.352	89.041	0.40 %
95	0.258	91.437	0.28 %
125	0.164	88.988	0.18 %
125	0.218	91.705	0.24 %

3.5 Comparison of Performance of R22 and R410A Systems

3.5.1 R410A Cooling Capacity Relative to R22

Cooling capacity comparisons were made between the R22 system and the R410A system with its different compressors by fitting a curve to the R22 cooling capacity as a function of outdoor temperature. All of the R22 cooling data points were fit to a polynomial function given below by equation 3.1:

$$Q(\text{Btu/h}) = 42196.9 - 4.8705\times10^{-3}\cdot T(^\circ F)^3 \quad (3.1a)$$

$$Q(\text{kW}) = 11.877 - 1.638\times10^{-5}\cdot T(^\circ C)^3 \quad (3.1b)$$

The cubic polynomial above fit the R22 capacity data with a Pearson correlation coefficient (R^2) of 0.978 and a fit standard error of 591.06 Btu/h (173.22 W) over the temperature range of 82.0 °F to 130.0 °F (27.8 °C to 54.4 °C). Using equation 3.1, the

R410A cooling capacity was divided by the R22 cooling capacity calculated at the appropriate outdoor temperature to calculate the cooling capacity ratio.

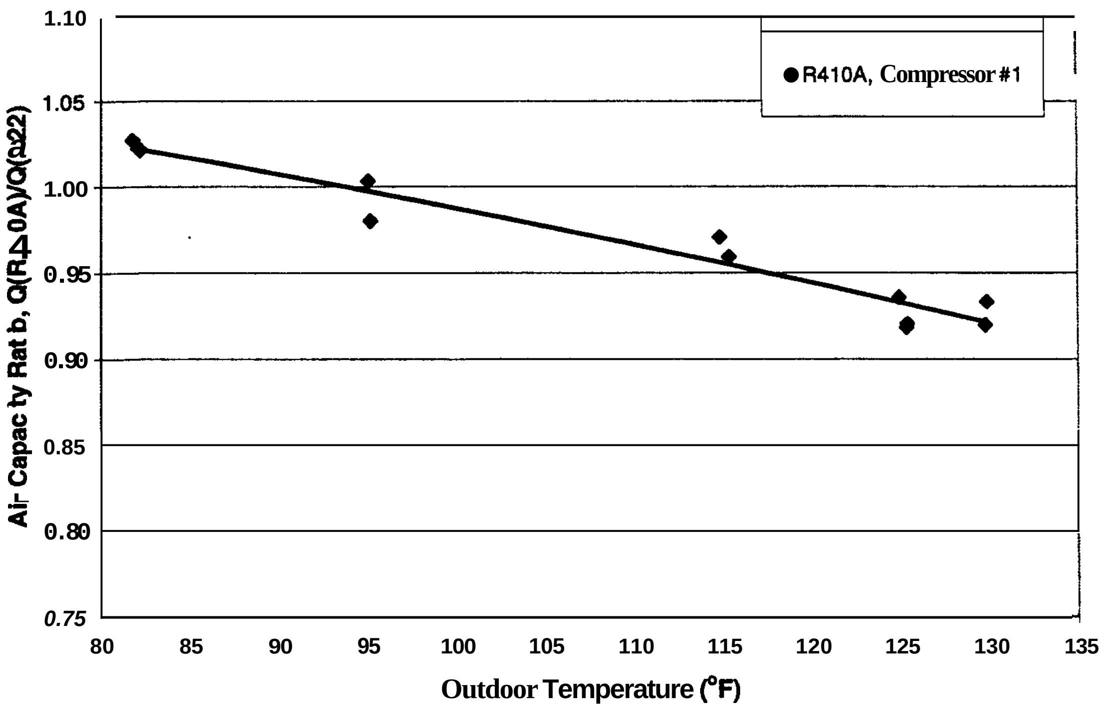


Figure 3.14 Cooling capacity of R410A system relative **to R22** system

The capacities of the R22 system and R410A system were equal at the 35.0 °C (95.0 °F) outdoor temperature. At the 82.0 "F (27.8 "C) rating point, the R410A system capacity was approximately 2 % greater than that of the R22 system. As the outdoor temperature increased, the capacity of the R410A system decreased more rapidly than the R22 system capacity, and at the 130.0 "F (54.4 "C) test point was 9 % below the R22 value.

3.5.2 R410A Cooling EER Relative to R22

Cooling EER for the R22 system was fit to a polynomial function of the outdoor temperature using statistical analysis software. Cooling EER (COP) as a function of outdoor temperature was fit to the polynomial shown below:

$$\mathrm{EER(Btu/Wh)} = 36.692 - 0.3611 \cdot T(°F)^{0.8981} \tag{3.2a}$$

$$\mathrm{COP} = 9.459 - 0.3323 \cdot T(°C)^{0.7654} \tag{3.2b}$$

The power law function fit R22 EER (COP) as a function of outdoor temperature with a Pearson correlation coefficient (R^2) of 0.998 and a fit standard error of 0.2749 Btu/Wh (0.081) over the outdoor temperature range of 82.0 °F to 130.0 °F (27.8 °C to 54.4 °C). Just as in the cooling capacity case, R410A EER (COP) was divided by the calculated R22 EER (COP) at the given outdoor temperature to produce a relative value.

The efficiency trend was similar to the trend for capacity; however, the performance degradation of R410A was more pronounced, as shown in Figure 3.15. At the 27.8 °C (82.0 °F) rating point, the EER (COP) of the R410A system was a few percent higher than that for the R22 system. At the 95.0 °F (35.0 °C) rating point, at which the capacities were equal, the R410A EER (COP) was approximately 4 % below the R22 EER (COP). At the highest ambient temperature of 130.0 °F (54.4 °C), the R410A EER (COP) was about 15 % lower than the EER (COP) of the R22 system. In addition to typical measurement uncertainties, we attribute the scatter shown in the EER (COP) ratios for a given outdoor temperature to day-to-day variations of voltage at the testing facility. However, these voltage variations were always within the range allowed by the **ARI** 210/240 **(1994).**

No rapid **drop** in capacity and **EER** occurred for the **R410A** system as the outdoor temperature increased to **155.0 °F (68.3 °C).** This is similar to the split-system testing performed by Wells et al. **1999.** They showed a 10 % lower capacity of the **R410A** system than the **R22** system at **125.0 °F (51.7 °C).** The literature has shown that capacity and EER **are** very sensitive to the expansion device and the size of the heat exchangers (Farzad and O'Neal **1991,** Gates et al. **1967).** This work has shown that **a TXV** in combination with sustained subcooling can mitigate some of the performance degradation seen at high ambient temperatures.

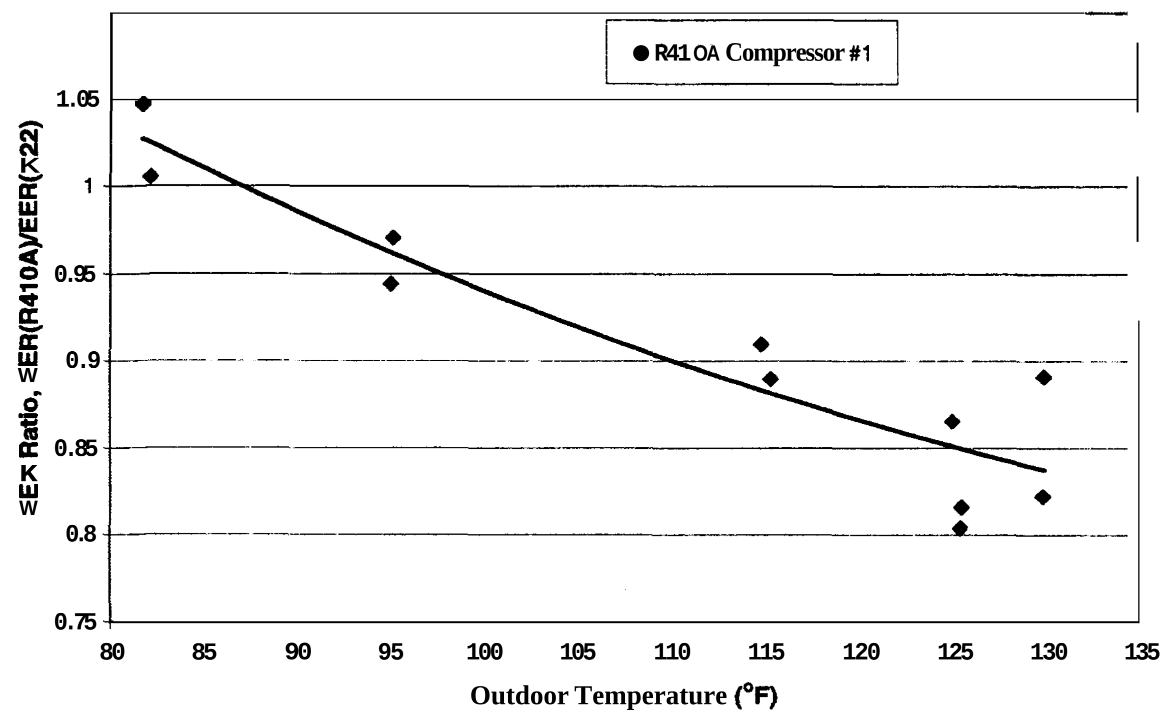


Figure **3.15** Cooling EER of **R410A** system relative to R22 system

4. MODELING

Laboratory tests conducted for this project provided performance information on **R22** and **R410A** systems and their components, including the evaporators and condensers. The two systems employed the evaporators and condensers of identical design, respectively. Hence, these data constitute a rare material for validation of simulation models working with different fluids.

Development of evaporator and condenser models is the dominant effort in building up a model of a vapor-compression air conditioner. This is because we can readily simulate compressor's performance using compressor maps available from the compressor manufacturer, and the expansion valve is an easy to model isenthalpic device. In fact, with the tube-by-tube representation **of** a finned-tube heat exchanger applied in the NIST heat pump model, ninety percent of the code is used for evaporator and condenser modeling. For this reason and considering the difficultly associated with designing optimized heat exchangers, our task included development and validation of the evaporator and condenser models, EVAPS and CONDS, and packaging them with a visual interface in one software package called EVAP-COND.

It is not in the scope of this report to provide a comprehensive description of the evaporator and condenser models but rather to provide practical information on how the models were formulated, what their features are, and how they can be **used.** This information is provided in sections **4.1** through **4.4.** The remaining sections discuss implementation of **EVAPS** and CONDS into a simulation model of an air conditioner, it's validation, and comparative simulations of R22, R410A, R134a, and R404A air conditioners.

4.1 Modeling Issues for Finned-tube Heat Exchangers

4.1.1 EVAP5 and **CONDS** Modeling Approach

Finned-tube air-to-refrigerant heat exchangers constitute the predominant heat exchanger type used in air conditioning. They are manufactured with a variety of refrigerant circuitry designs. Simulation models that account for refrigerant circuit architecture are better equipped for accurately predicting the heat exchanger performance. This is because the refrigerant path through the heat exchanger can have a significant effect on heat exchanger performance. The models presented here, EVAP5 and CONDS, originated in the tube-by-tube simulation model formulated by Chi (1979). Over the years, several significant new features were implemented, which included the capability to account for air maldistribution and its interaction with refrigerant distribution, extension of the models to zeotropic mixtures, extension to new refrigerant property representations, new simulation correlations, etc. (Domanski and Didion (1983), Domanski (1991), Lee and Domanski (1997), Domanski (1999b)).

Figure 4.1 presents the refrigerant circuitry and air velocity representation used by EVAP5 and CONDS. Due to the tube-by-tube modeling approach, the programs recognize each tube as a separate entity for which it calculates heat transfer. These calculations are based on inlet refrigerant and air parameters, properties, and mass flow rates. The simulation begins with the inlet refrigerant tubes and proceeds to successive tubes along the refrigerant path. At the outset of the simulation, the air temperature is only known for the tubes in the first row and has to be estimated for the remaining tubes. A successful run requires several passes (iterations) through the refrigerant circuitry, each time updating inlet air and refrigerant parameters for each tube.

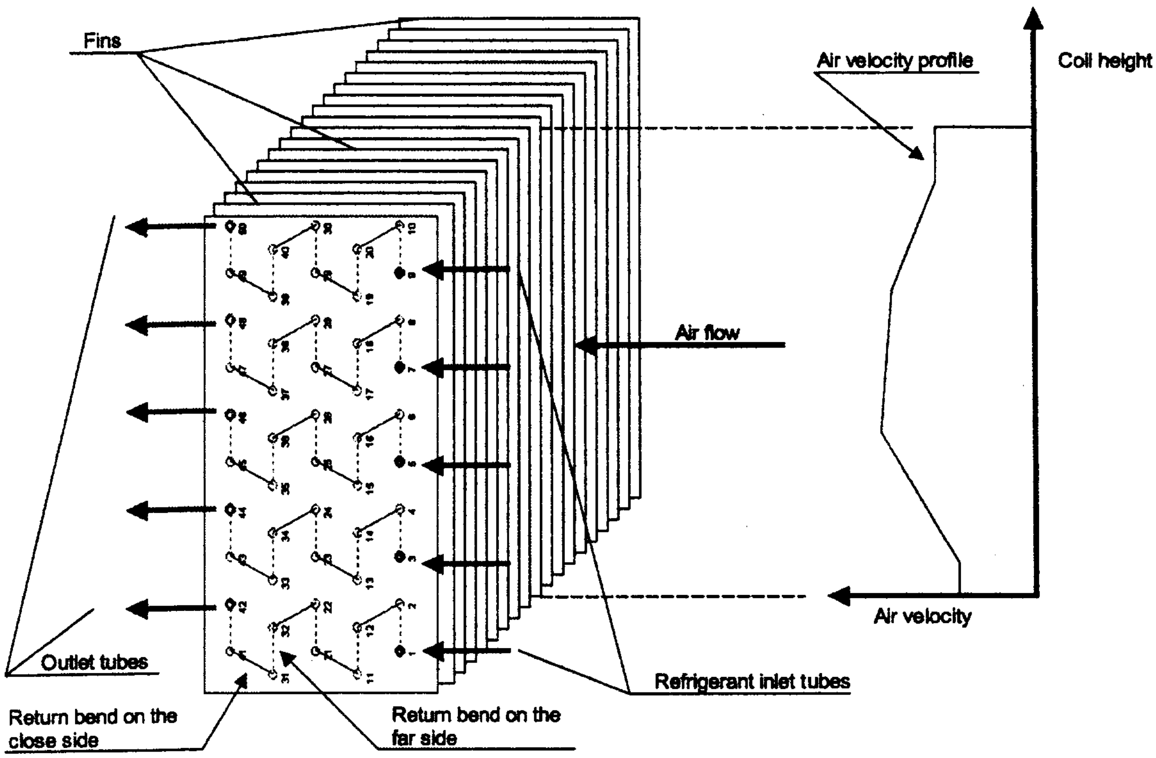


Figure **4.1** Representation of air distribution and refrigerant circuitry in **EVAPS** and **CONDS**

Heat transfer calculations start by calculating the heat-transfer effectiveness, **E**, by one of the applicable relations (Kays and London, **1984).** With the air temperature changing due to heat transfer, the selection of the appropriate relation for **E** depends on whether the refrigerant undergoes a temperature change during heat transfer. Once **E** is determined, heat transfer from air to refrigerant is obtained using eq. **(4.1).**

$$Q_r = m_a C_{pa} (T_{ai} - T_{ri}) \varepsilon \tag{4.1}$$

The overall heat-transfer coefficient, U, is calculated by eq. (4.2), which sums up the individual heat-transfer resistances between the refrigerant and the air.

$$U = \left(\frac{A_o}{h_i A_{pi}} + \frac{A_o X_p}{A_{pm} K_p} + \frac{1}{h_l} + \frac{A_o}{A_{po} h_{pf}} + \frac{1}{h_o (1+\alpha) \left(1 - \frac{A_f}{A_o} (1 - \varphi) \right)} \right)^{-1} \tag{4.2}$$

The first term of eq. **(4.2)** represents the refrigerant-side convective resistance. The second term is the conduction heat-transfer resistance through the tube wall, and the third term accounts for the conduction resistance through the water layer on the fin and tube. The fourth term represents the contact resistance between the outside tube surface and the fin collar. The fifth term is the convective resistance on the air-side where the multiplier $(1+\alpha)$ in the denominator accounts for the latent heat transfer on the outside surface. For a dry tube α = 0.0 and $1/h_l$ = 0.0. Once the heat transfer rate from the air to the refrigerant is calculated, the tube wall and fin surface temperatures can be calculated directly using heat-transfer resistances. Then, the humidity ratios for the saturated air at the wall and fin temperatures **are** calculated, and mass transfer from the moist air to the tube and fin surfaces is determined. For more detailed information on heat transfer calculations refer to Domanski (1991).

Simulating refrigerant distribution is an important part of simulating performance of heat exchangers, particularly for cases with non-uniform air distribution. In a heat exchanger with multiple circuits, refrigerant distributes itself in appropriate proportions so that the refrigerant pressure drop from inlet to outlet is the same for all circuits. This observation is essential for calculating a fraction of the total refrigerant mass flow rate flowing through a particular circuit. At the outset of this ARTI project, the evaporator model used a scheme for predicting refrigerant distribution that was based on the Pierre evaporation-pressure-drop correlation (Domanski, 1991), while the condenser model did not have this capability at all. Under the current project, we developed a new scheme for simulating refrigerant distribution between different circuitry branches by treating the problem as a nonlinear system of equations and solving it using the Newton-Raphson method. The new scheme can be applied to the evaporator, refrigerant distributor/evaporator system, and condenser regardless of the correlation used to calculate refrigerant pressure drop in a given heat exchanger. The Pierre-based method has been retained in EVAPS and CONDS to simulate refrigerant distribution for the first two iteration loops. At the outset of simulation, the initial refrigerant distribution is estimated based on the number of tubes in a given circuit and the circuit's layout (circuit split points and their locations).

4.1.2. Air-side Heat Transfer Correlations

A significant, often the major, part of heat transfer resistance between the air and refrigerant is on the air side of the heat exchanger. For this reason we reviewed literature on the latest air-side heat transfer correlations at the beginning of this project.

Figure 4.2 compares predictions of different correlation available in the literature. We calculated these predictions for typical designs for fins of different categories for a three-depth-row heat exchanger. The figure does not include a slit-wavy fin - the type used in the tested R22 and R410A systems - since we could not locate a slit-wavy fin correlation in the literature. The layout of different lines on the figure may serve as explanation why predicting performance of a finned-tube heat exchanger equipped with an enhanced fin may be difficult. For wavy fins, the correlation by Wang et al. (1999a) and Kim et al. (1997) are in a close agreement, while the correlation by Webb (1990) calculates heat transfer coefficients up to 50 % lower that the two first methods. At one point, the Webb correlation provides the wavy fin heat transfer coefficient to be lower than that for flat fins, which is not a realistic prediction.

For slit (lanced) fins, the correlations by Nakamura and Xu (1983) and Wang et al. (2001) are apart by as much as 40 %, depending on air velocity. This spread may be indicative of the general fact that some correlations do not predict well outside the geometries for which they were developed. A measurement uncertainty in one or both experiments may also be a contributing factor to the 40 % discrepancy. Regarding the louver fin, the correlation by Wang et al. (1999b) shows a step change caused by using two different algorithms depending on the Reynolds number. Two separate algorithms are also causing a step change in predictions by the Webb correlation for flat fins.

The analysis of relative predictions of air-side heat transfer coefficient for different surfaces prompted us to replace the existing correlations in EVAPS and CONDS with correlations published by Wang and his co-workers for all type of fins, i.e., flat, slit, wavy and louver fins. In our judgment, this will ensure a higher degree of prediction consistency when comparing performance of heat exchangers with different fin designs since it can be expected that a prediction consistency for the air-side heat transfer coefficient will be maintained by correlations developed by the same author. We may note one reservation with this choice; we expect a slit (lanced) fin to perform better in relation to a wavy fin than it is reflected by the slit and wavy Correlations authored by Wang (Wang et al., 2001, and Wang et al., 1999a). We could expect rather the values calculated by the Nakayama and Xu correlation would be in general more representative for most practical air velocities. However, we were disturbed by the fact that the Nakayama and Xu (1983) predictions do not degrade more at air velocities below 4 ft/s (the trend exhibited by the other correlations), and for this reason we opted for the Wang correlation for slit fins. Hence, the Wang correlation for slit fins (Wang et al., 2001) was selected not for its absolute predictions, but rather for the shape of the prediction line, which can be adjusted to proper absolute values with a correction multiplier.

All air-side heat transfer correlations authored by Wang include the tube-to-collar heat transfer resistance. Hence, his correlations do not have the ambiguity of the previously used flat-fin correlation by Gray and Webb (1986) developed using data on 16 heat exchangers, 10 of which had no fin-to-collar heat transfer resistance due to a metallurgical bond. For this reason, when incorporating the correlations by Wang et al. we removed the algorithm for calculating the tube-collar junction resistance, which up to this point was calculated by the correlation by Sheffield at al. (1988).

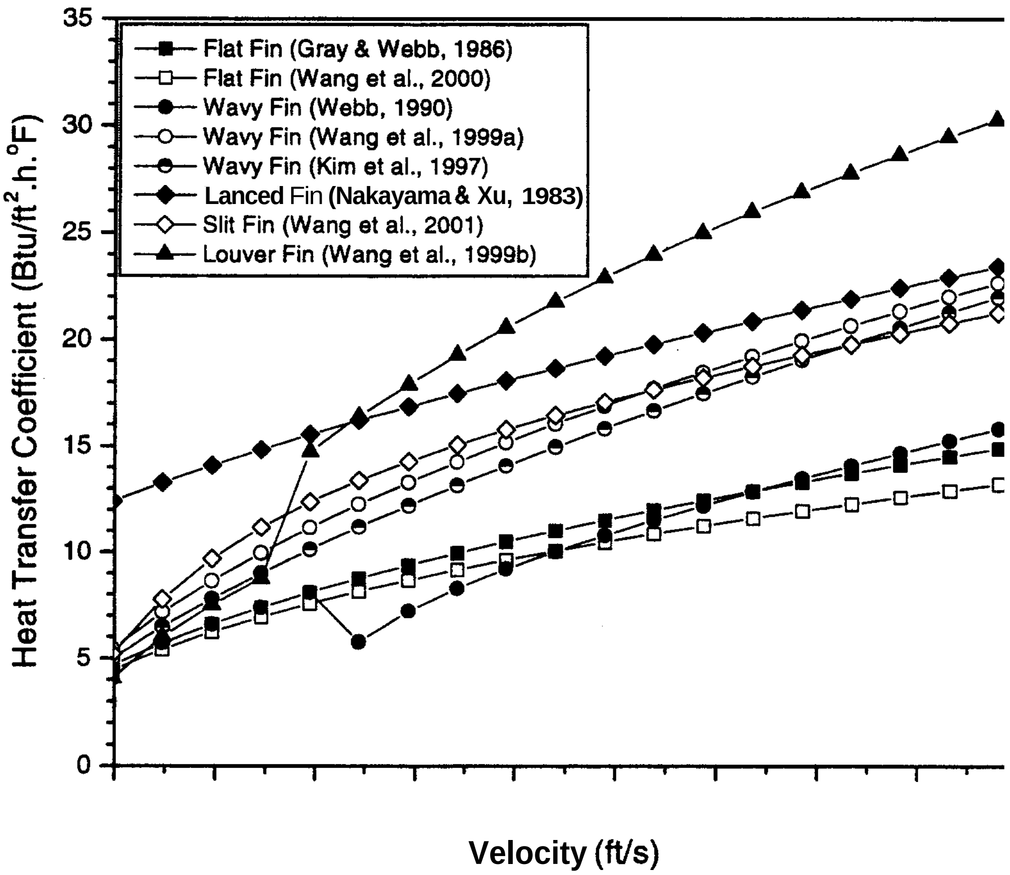


Figure **4.2** Comparison of air-side heat transfer correlations

4.1.3 Representation of Refrigerant Properties

The following upgrades were made to EVAP5 and CONDS under this project:

- Upgrade of **EVAPS** and **CONDS** to REFPROP6

  We upgraded EVAPS and **CONDS** from REFPROPS (Gallagher et al., 1996) to REFPROP6 (McLinden et al., 1998) routines. Since these refrigerant property packages employ different equations of state, we used a system of bridging routines to make the conversion.

- Development of Pressure-Enthalpy Based Property Look-up Tables

  EVAPS and CONDS simulations are computationally intensive, and using a refrigerant property look-up tables is a practical necessity if simulation runs are expected to last less than 60 seconds. This is particularly true in case of REFPROP6, for which property calculations **are** several times more CPU demanding than for REFPROPS. The pressure-quality-based look-up tables used so far with R22 were not adequate for this study since the R410A air conditioner may enter the transcritical operating regime at extremely high ambient temperatures. **A** new pressure-enthalpy based system of look-up tables was developed to facilitate simulation above the critical point of refrigerant. This look-up scheme includes eight different property routines that retrieve the desired state or transport property depending on the available properties identifying the refrigerant's thermodynamic state. The look-up scheme is applicable to single component refrigerants and refrigerant mixtures.

- Development of an Error Evasive Scheme for Refrigerant Property Calculations

  Although REFPROP6 provides improved representation of thermodynamic properties over REFPROPS, it occasionally crashes, particularly when calculating properties of refrigerant mixtures. Such occasional crashes may be acceptable in manually set-up property calculations with REFPROP's user's interface. However, they are unacceptable in heat exchanger and system simulations due to the high number of calls to property routines, which eventually may cause every simulation run to crash. To be able to use REFPROP6 routines, we developed an error evasive scheme that attempts to calculate properties even if REFPROP flash calculations do not converge, e.g., if PHFLSH crashes, a routine that uses TPRHO is invoked to attempt to iteratively match TPRHO's h value with the known (target) h value. If both REFPROP **flash** calculations do not converge, then the data in the refrigerant look-up table is flagged and look-up table routines iterate this point using refrigerant properties in the neighboring nodes of the table. If the critical pressure falls between the lower and upper pressure limits of the table, an additional set of data are generated for the critical pressure and the entire enthalpy grid (this is done to improve the accuracy of the iterations near the critical pressure).

4.2 Evaporator Model EVAP5

4.2.1 Heat Transfer and Pressure Drop Correlations

EVAPS uses the following correlations for calculating heat transfer and pressure drop.

Air Side

- heat-transfer coefficient for flat fins: Wang et al. (2000)
- heat-transfer coefficient for wavy fins: Wang et al. (1999a)
- heat-transfer coefficient for slit fins: Wang et al. (2001)
- heat-transfer coefficient for louver fins: Wang **et** al. (1999b)
- fin efficiency: Schmidt method, described in McQuiston et al. (1982)

Refrigerant Side

- single-phase heat-transfer coefficient, smooth tube: McAdams, described in **ASHRAE** (2001)
- evaporation heat-transfer coefficient up to 80% quality, smooth tube: Jung and Didion (1989)
- evaporation heat-transfer coefficient up to 80% quality, rifled tube: Jung and Didion (1989) correlation with a 1.9 enhancement multiplier suggested by Schlager et al. (1989)
- mist flow, smooth and rifled tubes: linear interpolation between heat transfer coefficient values for 80 % and 100 % quality
- single-phase pressure drop, smooth **tube:** Petukhov (1970)
- two-phase pressure drop, smooth tube, lubricant-free refrigerant: Pierre (1964)
- two-phase pressure drop, rifled tube: Pierre (1964) correlation for smooth tube with a 1.4 multiplier suggested by Schlager et al. (1989)

  We incorporated the correlation that accounts for a **0.5** % content of lubricant in the refrigerant.
- single-phase pressure drop, return bend, smooth tube: White, described in Schlichting (1968)
- two-phase pressure drop, return bend, **smooth** tube: Chisholm, described in Bergles et al. (1981)

  The length of a return bend depends on the relative locations of the tubes connected by the bend. **This** length was accounted for in pressure drop calculations.

4.2.2 EVAP5 Validation

We used evaporator test data obtained during R22 and R410A system tests to validate the evaporator model. For all tests, the indoor air dry-bulb temperature was 80.0 °F (26.7 °C) and the wet-bulb temperature was 67.0 "F (19.4 "C). The evaporator saturation temperature varied from 45.0 "F to 59.0 "F (7.2 °C to 15.0 "C) due to the wide range of outdoor temperature for which the R22 and R410A systems were tested (82.0 °F to 135.0°F (27.8 "C to 57.2 °C)). This resulted in different refrigerant inlet qualities, which ranged from 0.25 to 0.30.

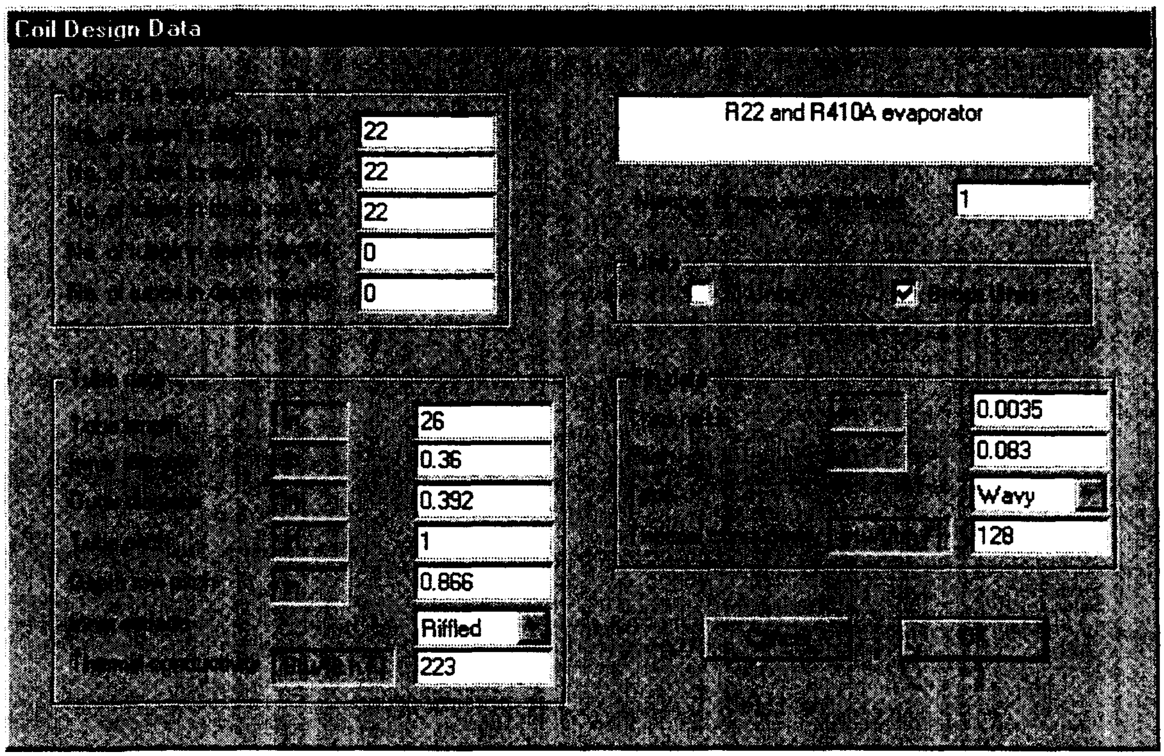


Figure 4.3 Design information for R22 and R410A evaporators

The R22 and R410A evaporators were identical heat exchangers. Copying from respective windows of the EVAP-COND interface, Figure 4.3 shows evaporator key design parameters, and Figure **4.4** shows a side-view schematic with the refrigerant circuit. The circles symbolize the tubes, the solid lines symbolize the returning bends on the near side, and the broken lines denote the returning bends on the far side. The refrigerant circuit had six branches. The refrigerant entered the evaporator via tubes 1, 13, 23, **35, 45, 57,** and left via tubes 12, 22, 34, **44,** 56, and **66.** The tube crossing over tube **28** and tube 38 is a result of graphical simplifications in the EVAP-COND interface. In fact, tube ***6*** feeds tube 51, tube 28 feeds tube 29, and tube ***50*** feeds tube 7.

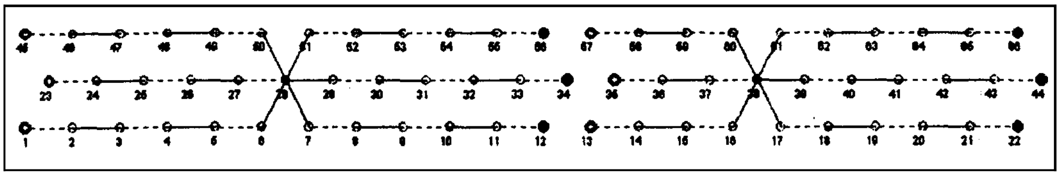


Figure 4.4 Refrigerant circuitry design in R22 and **R410A** evaporators

The evaporators were equipped with fins **of** a wavy/slit design. Since a correlation for **a** wavy/slit fin **is** not available in literature, we selected a wavy fin in our simulations and used a correction multiplier of 2.0 to compensate for the enhancement provided by the slits. This value approximate the average of the enhancements for a slit fin over a flat surface as calculated by Wang et al. (2001) and Nakayama and Xu (1983) for the air velocity range from **4 ft/s** to **5** ft/s (1.22 m/s to 1.52 m/s). If the Nakayama **and Xu** correlation was selected for our simulations, the same value of the air-side heat transfer coefficient would be obtained by applying a correction multiplier of 0.30 to represent the enhancement due to the wavy design. **As** we note that this correction is lacking analytical rigor, we also have to recognize that not providing any correction is not proper **as** well.

For our validations, we selected five **R22** test points and seven **R410A** data points. The **R22** data points were obtained during system tests at **82.0** "F **(27.8 "C), 95.0 °F (35.0 °C), 115.0** "F **(46.1 "C), 125.0 °F (51.7 "C),** and **135.0 °F (57.2 "C)** ambient temperatures. The **R410A** data points came from the **82.0** "F **(27.8 °C), 95.0 °F (35.0 °C), 115.0 °F (46.1 "C), 125.0** "F **(51.7 °C),** and **130.0 °F (54.4 "C)** tests using the original compressor, and the **150.0** "F **(65.6 "C)** and **155.0** "F **(68.3 "C)** tests using the custom-fabricated compressor. The inlet operating parameters included the air dry-bulb temperature, relative humidity, air volumetric flow rate, refrigerant inlet quality (based on refrigerant thermodynamic state at the TXV inlet), and refrigerant saturation temperature and superheat at the evaporator outlet. The program iterated inlet pressure and refrigerant mass flow rate to obtain the target conditions at the evaporator outlet.

Tables **4.1** shows the refrigerant input parameters and simulation results for the **R22** evaporator, while Figures **4.5** and **4.6** graphically present evaporator total capacities and sensible heat ratios. For **R22,** the inlet quality ranged from **0.15** to **0.34,** and the saturation temperature for different tests was between **48.7** "F **(9.3 "C)** and **53.9 °F (12.2 "C).** These variations resulted in a capacity difference between different tests being **as** much **as 30** %. **EVAP5** underpredicted all measured capacities with three underpredictions being within the **3.7 %** and **4.7 %** range and two significantly larger underpredictions of **7.4** % and **11.4** %. **EVAPS** overpredicted the sensible heat ratio by **6.1** % to **10.7** %.

Table **4.2** shows the refrigerant input parameters and simulation results for the R410A evaporator, while Figures **4.7** and **4.8** graphically present evaporator total capacities and sensible heat ratios. For the seven **R410A** tests, the evaporator saturation temperature was between 50.5 **°F (10.3 °C)** and **59.1 °F (15.1 °C),** and the inlet quality ranged from **2.0** % to **4.7** %. This range **was** greater for R410A than for **R22.** In general, a higher inlet quality for R410A than that for **R22** was a result of a lower R410A critical temperature. The highest value of inlet quality came from the test at the **155.0 °F (68.3 °C)** ambient temperature, to which the **R22** system was not subjected. **As** a result of different operating conditions, evaporator capacities differed by **as** much as **46** %. Table **4.2** shows that **EVAP5** predictions were more consistent with R410A than with **R22.** For tests with evaporator saturation temperatures up to 55.0 **°F (12.8 °C)** (practical operation range), **EVAPS** underpredictions were approximately **4.5** % **± 1.3** %. **At** higher saturation temperatures, evaporator total capacities were within **2.7** % of the tested value. **As** was the case with **R22** predictions, the model overpredicted the sensible heat ratio for all tests by a similar percentage.

EVAPS provides detail information on performance of individual tubes. Predictions of refrigerant superheat (or quality and temperature) for evaporator exit tubes are indicative of the correctness of this information. Tables **4.3** and **4.4 show** exit temperatures, qualities, and refrigerant distribution for **R22** and R410A evaporators. The tube number identifiers used are consistent with Figure **4.4.** The level of agreement in outlet temperatures between the tested and simulated values is better for the **R22** evaporator than for **the R410A** evaporator due to the higher overall superheat set in the **R22** evaporator. (In an extreme case with an undercharged system, the refrigerant exit temperature approaches the *air* temperature). Considering the difficulty of predicting individual superheats, the simulated values can be considered **as** acceptable.

A common aspect of **R22** and **R410A** predictions is the underprediction of refrigerant superheat in tube **44.** In EVAPS, this underprediction is caused by a lower air mass flow rate seen by tube **44** because this tube is located next to the side of the heat exchanger. It appears that in a real heat exchanger, heat transfer via fins between neighboring tubes provides a compensating effect. This heat transfer has been neglected in EVAPS and should be studied for proper implementation into the next version of the model.

The observed capacity underpredictions for **R22** and **R410A** evaporator were reasonably consistent, and improved prediction accuracy could be obtained by tuning the evaporator model farther. While accurate simulation of the evaporator is important, we should note that inaccurate predictions of evaporator capacity are "scaled down" when the evaporator model is incorporated into a model of an *air* conditioner. According to a simplified algorithm (Domanski, 1990), a 10 % lower evaporator capacity will result in a 3.6 % lower cooling capacity of the system. This is due to the fact that the system will rebalance itself at different saturation temperatures in the evaporator and condenser when inaccurate heat exchanger predictions are generated.

Table 4.1 EVAP5 validation with R22 evaporator

Test number	Refrigerant input data			Test results			Simulation results			Difference*	
	Inlet quality (fraction)	Outlet Tsat (°F)	Outlet Tsup (°F)	Total capacity (Btu/h)	Sensible capacity (Btu/h)	Sensible heat ratio (fraction)	Total capacity (Btu/h)	Sensible capacity (Btu/h)	Sensible heat ratio (fraction)	Total capacity (%)	Sensible heat ratio (%)
1208a	0.15	48.7	13.7	39364	28892	0.73	37925	29533	0.78	-3.7	6.1
10105	0.20	50.0	11.5	38640	28306	0.73	35777	28414	0.79	-7.4	8.4
010108a	0.22	51.6	10.3	34782	26784	0.77	33154	27557	0.83	-4.7	7.9
010110a	0.31	52.7	9.8	33420	25659	0.77	29617	25250	0.85	-11.4	10.7
1212a	0.34	53.9	11.6	30270	24808	0.82	28899	25510	0.88	-4.5	7.7

*100% (simulated value – tested value)/tested value

Table 4.2 EVAP5 validation with R410A evaporator

Test number	Refrigerant input data			Test results			Simulation results			Difference*	
	Inlet quality (fraction)	Outlet Tsat (°F)	Outlet Tsup (°F)	Total capacity (Btu/h)	Sensible capacity (Btu/h)	Sensible heat ratio (fraction)	Total capacity (Btu/h)	Sensible capacity (Btu/h)	Sensible heat ratio (fraction)	Total capacity (%)	Sensible heat ratio (%)
b010330k	0.20	50.5	4.5	40345	29412	0.73	39072	29933	0.77	-3.2	5.1
b010328a	0.25	52.1	3.8	38144	27998	0.73	36509	28772	0.79	-4.3	7.4
b010331x	0.29	54.4	3.5	33305	26677	0.80	31396	27433	0.87	-5.7	9.1
010403a	0.33	55.2	3.9	30586	25356	0.83	29262	26340	0.90	-4.3	8.6
b010425x	0.35	56.0	4.2	29414	24884	0.85	28611	25660	0.90	-2.7	6.0
C010719a	0.44	58.1	5.7	24801	22519	0.91	24894	23748	0.95	0.4	5.1
C010723d	0.47	59.1	5.3	22645	22590	1.00	22916	22916	1.00	1.2	0.2

* 100% (simulated value – tested value)/tested value

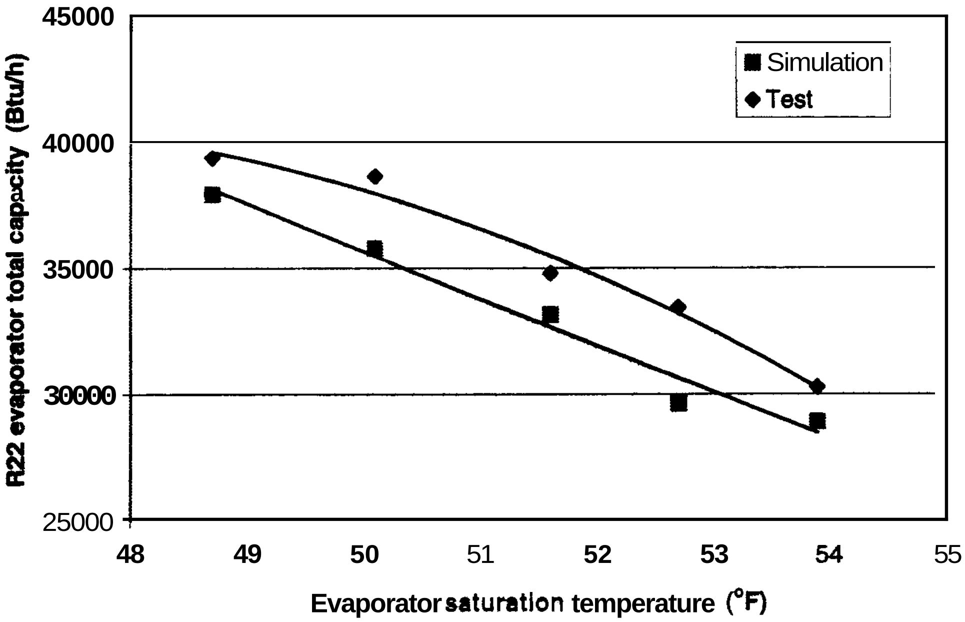


Figure **4.5** Tested and predicted capacities for **R22** evaporator

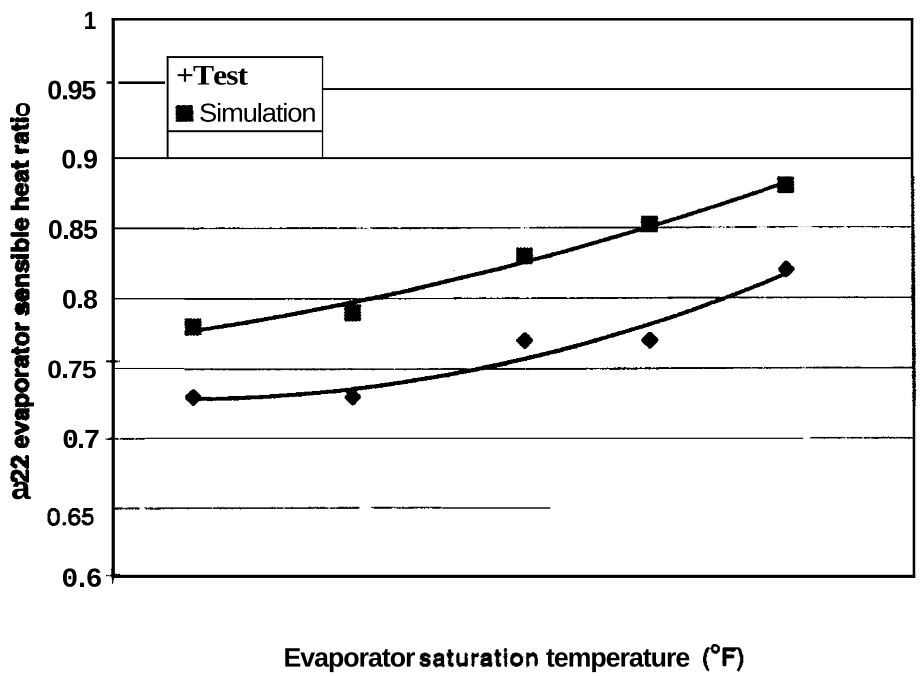


Figure **4.6** Tested and predicted sensible heat ratios for **R22** evaporator

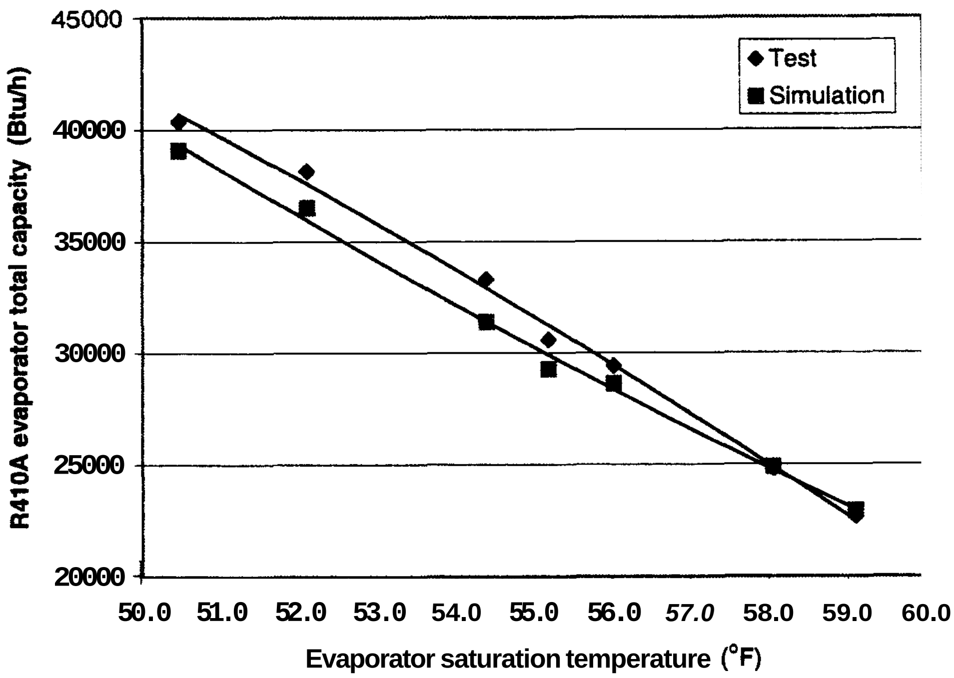


Figure 4.7 Tested and predicted capacities for R410A evaporator

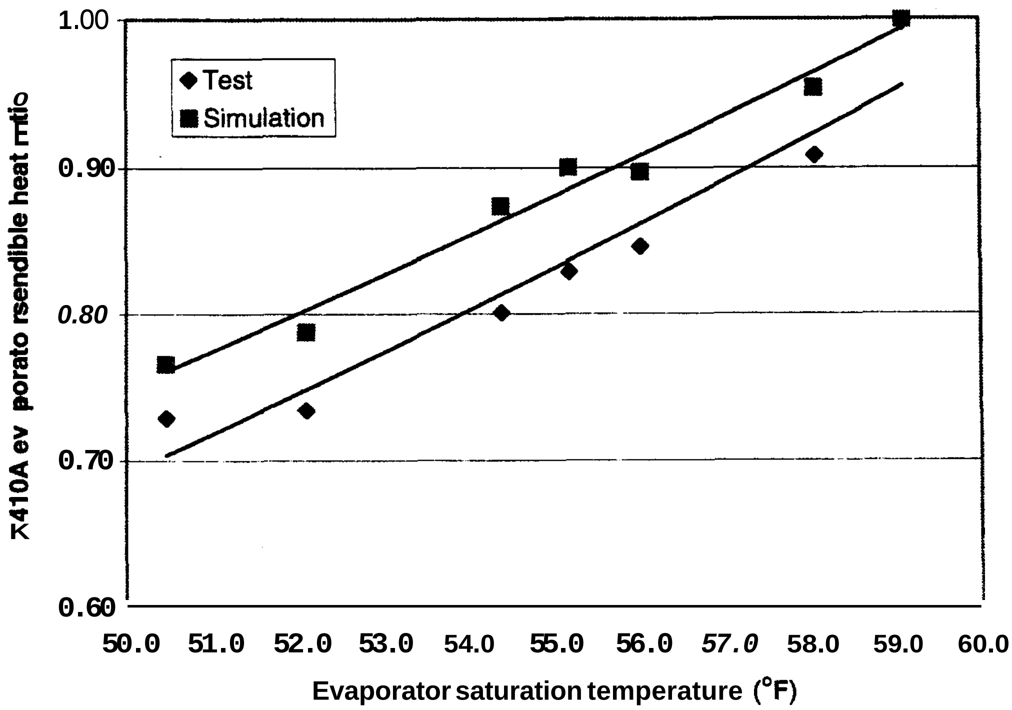


Figure 4.8 Tested and predicted sensible heat ratios for R410A evaporator

Tube Number	Test	Simulation		
	Outlet temperature (°F)	Outlet temperature (°F)	Outlet quality (fraction)	Refrigerant distribution (fraction)
12	63.3	74.5	1.00	0.167
22	63.5	71.0	1.00	0.168
34	62.5	63.1	1.00	0.166
44	58.5	51.3	1.00	0.169
56	60.6	63.2	1.00	0.163
66	55.6	53.6	1.00	0.166

Tube Number	Test	Simulation		
	Outlet temperature (°F)	Outlet temperature (°F)	Outlet quality (fraction)	Refrigerant distribution (fraction)
12	70.2	70.7	1.00	0.167
22	64.9	65.6	1.00	0.168
34	71.3	55.8	1.00	0.166
44	72.8	50.5	0.94	0.169
56	69.8	60.4	1.00	0.163
66	70.4	50.5	0.98	0.167

4.3 Condenser Model COND5

4.3.1 Heat Transfer and Pressure Drop Correlations.

COND5 uses the following correlations for calculating heat transfer and pressure drop.

Air Side

- heat-transfer coefficient for flat fins: Wang et al. (2000)
- heat-transfer coefficient for wavy fins: Wang et al. (1999a)

- heat-transfer coefficient for slit fins: Wang et al. (2001)
- heat-transfer coefficient for louver fins: Wang et al. (1999b)
- fin efficiency: Schmidt method, described in McQuiston et al., (1982)

Refrigerant Side

- single-phase heat-transfer coefficient, smooth tube: McAdams, described in ASHRAE (2001)
- condensation heat-transfer coefficient, smooth tube: Shah (1979)
- condensation heat-transfer coefficient, rifled tube: Shah (1979) correlation with a 1.9 enhancement multiplier suggested by Schlager et al. (1989)
- single-phase pressure drop, smooth tube: Petukhov (1970)
- two-phase pressure drop, smooth tube: Lockhart and Martinelli (1949)
- two-phase pressure drop, rifled tube: Lockhart and Martinelli (1949) correlation for smooth tube with a 1.4 multiplier suggested by Schlager et al. (1989)

  We incorporated the correlation that accounts for a 0.5 % content of lubricant in the refrigerant.
- single-phase pressure drop, return bend, smooth tube: White, described in Schlichting (1968)
- two-phase pressure drop, return bend, smooth tube: Chisholm, described in Bergles et al. (1981)

  The length of a return bend depends on the relative locations of the tubes connected by the bend. This length **was** accounted for in pressure drop calculations.

4.3.2 **CONDS** Validation

To validate the condenser model, we applied condenser test data obtained during the same R22 and R410A system tests we used for EVAPS validation. The R22 and R410A condensers were identical heat exchangers. Copying from respective windows of the EVAP-COND interface, Figure **4.9** shows condenser key design parameters, **and** Figure **4.10** shows a side-view schematic with the refrigerant circuit. **As** for the evaporator, the circles symbolize the **tubes,** the solid lines symbolize the returning bents on the near side, and the broken lines denote the returning bends on the far side. The refrigerant circuit had four branches merging into tube **25.** The refrigerant entered the condenser via tubes **32, 33, 44,**

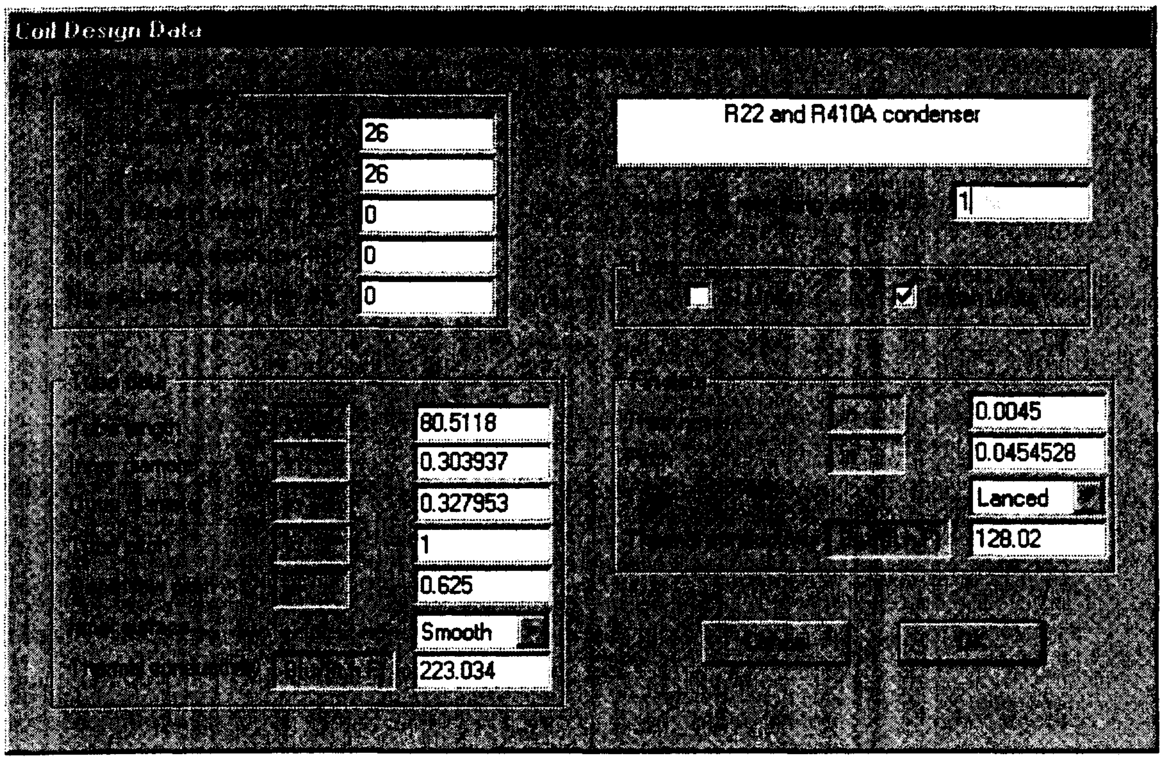


Figure *4.9* Design information for R22 and R410A condensers

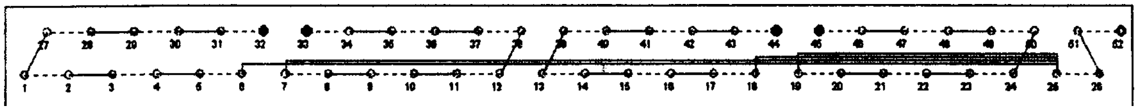


Figure **4.10** Refrigerant circuitry design in **R22** and **R410A** condensers

and **45,** and left via tube **52.** The refrigerant circuit shown in Figure **4.10** (and simulated **by COND5)** is a modified version of the circuit implemented in the tested condensers. The difference is in the final tubes in that the actual condenser had tubes **51** and **52** placed in the first depth row extending the condenser to the right side of tube **26,** and tube spaces in the second depth row were empty. (Figure **3.1** shows the actual refrigerant circuitry.) The current version of COND5 cannot handle "non-existing"tubes ***so*** this circuitry simplification was necessary. Considering the large size (for cooling capacity) of this condenser and its low refrigerant/air approach temperature, we believe that our simplification of circuitry does not compromise **CONDS** simulation to a significant degree.

The **R22** data points were obtained during system tests at **82.0** "F **(27.8** "C), **95.0** "F **(35.0 °C), 115.0** "F **(46.1 "C), 125.0** "F **(51.7 "C),** and **135.0** "F **(57.2 "C)** ambient temperatures. The **R410A** data points came from the **82.0** "F **(27.8 "C),** 95.0 "F **(35.0** "C), **115.0 "F (46.1 "C), 125.0** "F **(51.7** °C), and **130.0** "F **(54.4 "C)** tests using the original compressor and the **150.0 °F** (65.6 "C) and **155.0** "F **(68.3** "C) tests using the custom-fabricated compressor. The condensers wavy fins were tightly spaced **(22** fins per inch **(9** fins/cm)). Simulation input parameters included the air dry-bulb temperature, relative humidity, air volumetric flow rate, and refrigerant inlet temperature and pressure.

Table **4.5** shows the input parameters and simulation results for the **R22** condenser, while Figures **4.11** and **4.12** graphically present the condenser capacities and pressure drops at different ambient temperatures. We selected the ambient air temperature **as** the abscissa because it provides a common reference scale for **R22** and R410A test points. **CONDS** predicted condenser capacities well, within **1.6 % of** the measured values; however, underpredicted refrigerant pressure drops by **32.3 %** to **55.7 %**.

Table **4.6** shows the refrigerant input parameters and simulation results for the R410A condenser, while Figures **4.13** and **4.14** depict the condenser capacities and pressure drops. COND5 overpredicted capacities for all tests. The largest overprediction was **3.6** %. Pressure drop predictions were acceptable for the normal operating range **(82.0 °F (27.8 °C)** to **95.0 °F (35.0 °C)** ambient temperature). At higher temperatures, these underpredictions increased to above 50 % when the condenser operated close and above the R410A critical pressure.

While the reported capacity predictions were satisfactory, we have to note that these tests do not conclusively validate the condenser model because of the low approach temperature between the air and refrigerant during the condenser tests (maximum **6.9 °F (3.8** °C)). Since the tested condensers operated near a temperature pinch, additional tests with larger approach temperatures condensers would provide needed data to **fully** validate **CONDS.**

Table 4.5 COND5 validation with R410A condenser

Test number	Air dry-bulb temperature (°F)	Refrigerant input data					Test results		Simulation results		Difference*	
		Condenser inlet		Condenser outlet		rmass	Capacity	Pressure drop	Capacity	Pressure drop	Capacity	Pressure drop
		Temperature (°F)	Pressure (psia)	Temperature (°F)	Pressure (psia)	(lb/h)	(Btu/h)	(psi)	(Btu/h)	(psi)	(%)	(%)
1208a	81.7	150.0	217.3	88.6	197.5	534.0	46351	19.8	46953	13.4	1.3	-32.3
10105	95.1	168.4	259.1	100.0	240.5	537.6	45911	18.6	46618	12.4	1.5	-33.3
010108a	115.1	199.8	326.4	119.6	307.4	529.8	44185	19.0	44434	9.8	0.6	-48.4
010110a	125.4	218.6	365.9	129.9	346.9	525.6	43520	19.0	44204	9.8	1.6	-48.4
1212a	135.0	237.2	399.0	141.7	378.7	513.6	41961	20.3	42380	9.0	1.0	-55.7

*100% (simulated value – tested value)/tested value

Table 4.6 COND5 validation with R410A condenser

Test number	Air dry-bulb temperature (°F)	Refrigerant input data					Test results		Simulation results		Difference*	
		Condenser inlet		Condenser outlet		rmass	Capacity	Pressure drop	Capacity	Pressure drop	Capacity	Pressure drop
		Temperature (°F)	Pressure (psia)	Temperature (°F)	Pressure (psia)	(lb/h)	(Btu/h)	(psi)	(Btu/h)	(psi)	(%)	(%)
b010330k	82.1	141.4	341.6	89.5	329.1	537.0	47525	12.5	48861	11.4	2.8	-8.4
b010328a	95.0	159.2	399.8	100.7	388.8	541.8	46703	11.1	47743	10.7	2.2	-3.2
b010331x	115.4	190.9	507.1	122.3	497.0	539.4	43961	10.2	45538	8.4	3.6	-17.2
b010403a	125.0	208.8	559.1	130.2	549.6	525.0	42683	9.4	43841	7.6	2.7	-19.8
b010425x	129.9	218.6	587.4	134.7	578.0	521.4	42233	9.4	43313	7.2	2.6	-23.4
c010719a	150.0	248.6	709.1	152.3	699.4	517.2	39618	9.7	40170	4.4	1.4	-54.4
c010723d	155.4	261.6	742.8	156.9	733.5	495.6	37913	9.3	38039	4.5	0.3	-51.1

*100% (simulated value – tested value)/tested value

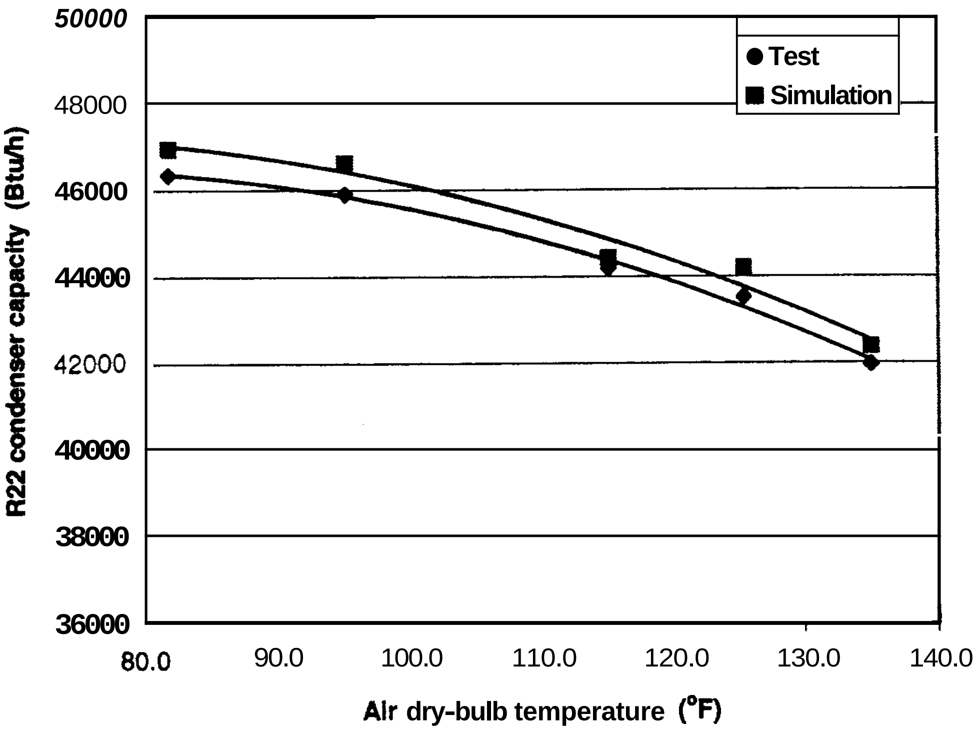


Figure 4.11 Tested and predicted capacities for R22 condenser

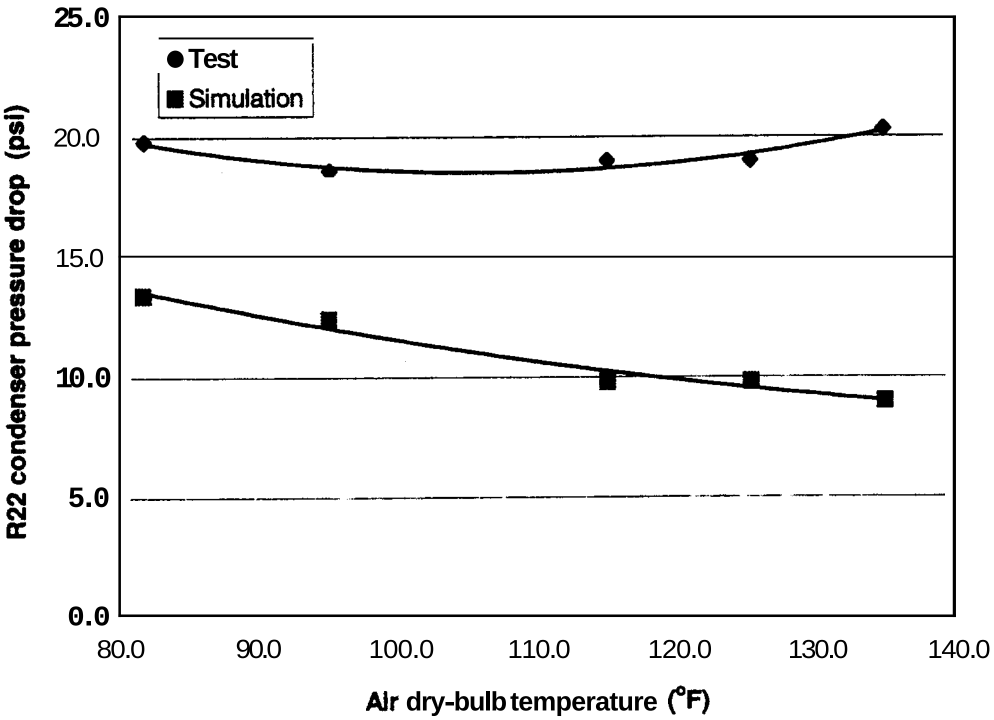


Figure 4.12 Tested and predicted pressure drops for R22 condenser

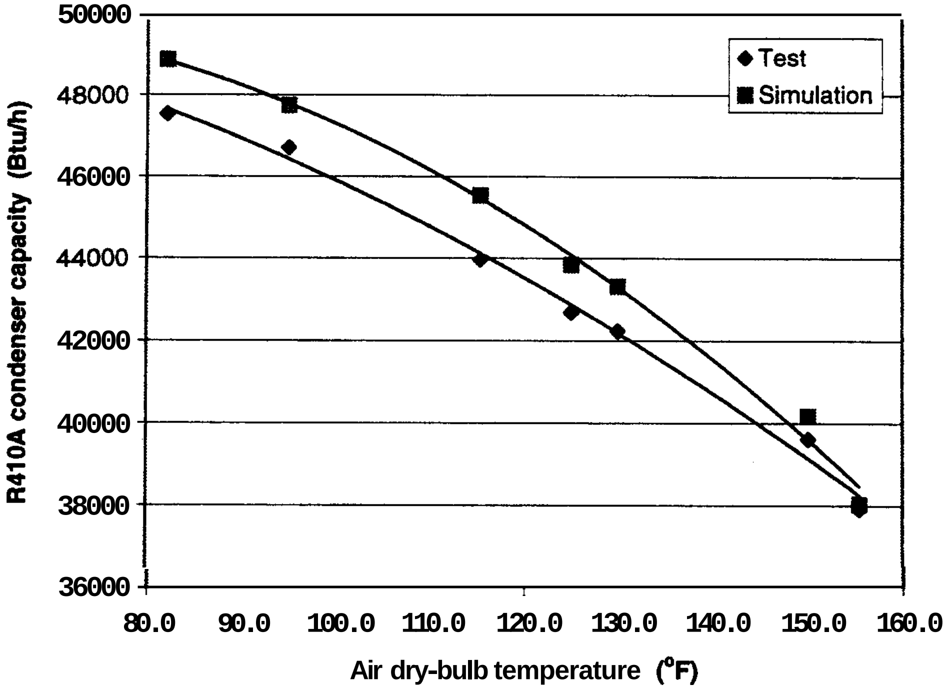


Figure 4.13 Tested and predicted capacities for R410A condenser

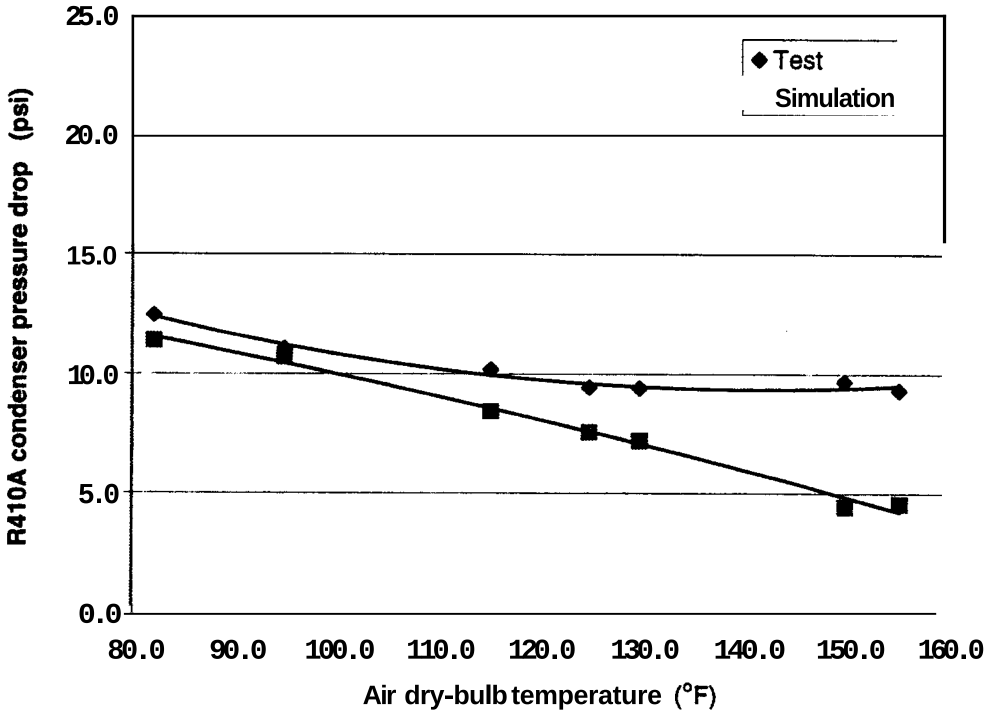


Figure 4.14 Tested and predicted pressure drops for **R410A condenser**

4.4 EVAP-COND Simulation Package

This project built on the ongoing effort at NIST to develop a simulation package for finned-tube evaporators and condensers called EVAP-COND. The beta version of EVAP-COND package is attached to this report. Version 1 of EVAP-COND is scheduled to be available for free download from http://www.bfrl.nist.gov/863/refrig.html in November 2002.

At the outset of this project, the first version of the graphical user interface **(GUI)** was already developed and linked with the evaporator model, EVAPS, allowing evaporator simulations using two options for input data. The effort exerted under this project led to new capabilities for EVAP-COND. Some of the new features go beyond the scope of the current project and were implemented due to supplemental support from NIST and DOE. The major upgrades of EVAP-COND are:

- Implementation of a refrigerant selection option and implementation of ten refrigerants (ammonia, propane, carbon dioxide, R134a, R22, **R32,** R407C, R404A, R507A, R410A).
- Implementation of corrections parameters for air-side heat transfer coefficient, refrigerant heat transfer Coefficient, and refrigerant pressure drop.
- Implementation of REFPROP6-based refrigerant properties
- Implementation of six new input data options for simulation of the evaporator. (Total of eight options are available.) The new options include simulation of the evaporator in conjunction with the refrigerant distributor for improved simulation of refrigerant distribution. A nominal pressure drop in the distributor lines is assumed.
- Incorporation of the condenser simulation model CONDS, which simulation capabilities include:

- subcritical and supercritical operation
- non-uniform air distribution
- refrigerant distribution

The scope of this project did not include development of a User's Manual. To facilitate the use of EVAP-COND, we prepared visual instruction pages in lieu of the manual. These pages are presented in Appendix D.

4.5 Modeling of Air Conditioner

4.5.1 Structure of Simulation Model

We formulated a simulation model of a vapor-compression air conditioner equipped with a TXV as the expansion device to simulate the systems tested under this project. The model consists of the following models of system components: evaporator, suction line, compressor, discharge line, and condenser (see Figure 4.15). The liquid line is not modeled; the practical significance of this simplification is that heat transfer between the ambient and liquid line is not accounted for. Physical modeling of the TXV is substituted with the assumption of a constant refrigerant superheat at the evaporator outlet and a constant refrigerant subcooling at the condenser outlet. Our test data show that the later assumption is less rigorous then the assumption of a constant superheat, however, it appears to facilitate accurate performance predictions better than using other simulation constraints (e.g. refrigerant mass inventory). For the two R410A simulations that involved transcritical operation at the ambient temperatures of 150.0°F (65.6 °C) and 155.0°F (68.3 °C) we used a constraint of specified approach temperature in place of a constant subcooling at the condenser outlet.

The condenser and evaporator models are those incorporated in the EVAP-COND package. The compressor is represented by a compressor map routine implementing a ten-term correlation described in ARI Standard 540 *(ARI* 1999). A correction is provided for a different than tested vapor superheat at the compressor inlet. The suction and discharge line model accounts for refrigerant pressure drop and heat transfer between the refrigerant and ambient. Most of the support routines have their origin in the NIST HPSIM simulation model (Domanski and Didion, 1983). They were either applied directly or modified. All refrigerant properties are calculated using REFPROP6 (McLinden et al., 1998) refrigerant property look-up tables. Having tested the program with REFPROPS (Gallagher et al., 1996) and REFPROP6, we saw no practical differences in predictions with R22 and visible differences with R410A. The formulated air conditioner model consists of 109 routines (93 subroutines and 16 functions) with 90% of them supporting simulations of the evaporator and condenser. The total count of subroutines does not include neither the REFPROP6 code nor the visual interface routines.

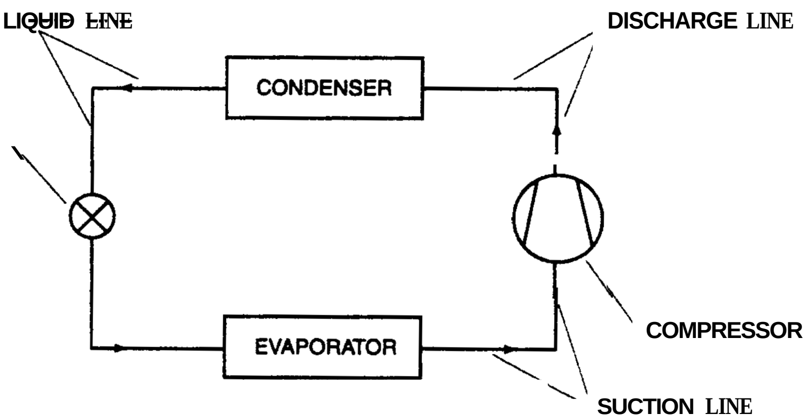


Figure 4.15 Component schematic of a tested air conditioner

4.5.2 Model Validation with R22 and R410A Air Conditioner Test Data

We validated the AC model with the five R22 and seven R410A system test data we used for EVAP5 and COND5 validations. For R22, these test points were obtained at outdoor temperatures of 82.0 "F (27.8 "C), 95.0 °F (35.0 "C), 115.0 °F (46.1 "C), 125.0 "F (51.7 "C), and 135.0 "F (57.2 "C). For R410A, the points taken at 82.0 "F (27.8 "C), 95.0 "F (35.0 "C), 115.0 "F (46.1 "C), 125.0 "F (51.7 "C), and 130.0 "F (54.4 "C) were obtained with the original compressor, while the tests at 150.0 "F (65.6 "C) and 155.0 "F (68.3 "C) were obtained using a custom-fabricated compressor of similar characteristics but a different electric motor. We included the 150.0 °F (65.6 "C) and 155.0 °F (68.3 "C) conditions with the understanding that these tests cannot be accepted as rigorous validation points.

When performing simulations at different operating conditions, we used as input the same evaporator superheat and condenser subcooling as it was measured during the 95.0 "F (35.0 "C) outdoor temperature test. This approach validates the adequacy of the assumption of constant superheat and subcooling for modeling TXV-equipped systems. All simulation runs were performed without making any adjustments to the simulation model beyond the adjustment for the evaporator air-side heat transfer coefficient discussed in chapter 4.2.2.

Tables 4.7 presents selected validation results for the R22 system. Figures 4.16 and 4.17 graphically present capacities and EERs at different operating conditions. The model predicted capacities within 1.9 % for all tests except the test at 135.0 "F (57.2 "C), where the simulated and tested capacities differed by 6.3 %. EER at the 135.0 °F (57.2 "C) test point also has the largest prediction error of 7.8 %. Other EERs are predicted within 5.1 %. We may note that assuming a constant evaporator superheat and condenser subcooling is not responsible for the highest prediction error at the 135.0 °F (57.2 "C) outdoor temperature; a simulation run performed with the tested values of superheat and subcooling yielded a capacity that was within 100 Btu/h and EER that was within 0.1 of the simulated values reported in Table **4.7.**

Table **4.8** presents validation results for the R410A system, and Figures 4.18 and 4.19 compare tested and simulated capacities and EERs, respectively. While predictions **are** very good for the low-end ambient temperatures (1**15.0 °F** (46.1 °C) and lower), their accuracy tends to deteriorate when the ambient temperature is above 115.0 **°F** (46.1 "C), **as** this also was the case with the R22 system. We can connect this trend with improper predictions of refrigerant mass flow rate. Figure 4.19 displays this disparity for the R410A system; a similar disparity took place for the R22 system **as** well. Since validations **of** EVAP5 and **CONDS** did not show significant disparity between simulation and test results, detailed examination of the compressor performance and its representation in our system model is merited in the future. At this point it can be speculated that the increase in refrigerant mass **flow** simulated by compressor map correlations was in response to a higher suction pressure (and higher density of the suction vapor), which dominated the effect of the lower compressor volumetric efficiency at **an** increased compressor pressure ratio. The difference between compressor map correlations and a real system is that compressor map tests were performed at 90.0 **°F (32.2** °C), while during system tests the compressor **was** exposed to the ambient temperature. At high ambient temperatures heat transfer from the compressor to the ambient **was** inhibited, which may have resulted in significant internal heat transfer and large refrigerant superheat at the suction valve and led to a decrease in refrigerant mass **flow** rate **as** compared to compressor map predictions.

Table 4.7 AC model validation with R22 system data

Test number	Outdoor dry-bulb temperature (°F)	Test results						Simulation results						Difference*	
		Capacity (Btu/h)	Work (W)	EER (Btu/h.W)	rmass (lb/h)	Evap. outlet pressure (psi)	Cond. inlet pressure (psi)	Capacity (Btu/h)	Work (W)	EER Btu/h.W)	rmass (lb/h)	Evap. outlet pressure (psi)	Cond. inlet pressure (psi)	Capacity (%)	EER (%)
1208a	81.7	39364	2225	17.7	534.0	96.6	217.3	39729	2283	17.4	525.0	93.1	213.0	0.9	-1.6
10105	95.1	38640	2529	15.3	537.6	98.8	259.1	37905	2614	14.5	528.0	94.6	251.7	-1.9	-5.1
010108a	115.0	34782	3221	10.8	529.8	101.4	326.4	35026	3100	11.3	534.0	97.2	318.4	0.7	4.6
010110a	125.4	33421	3673	9.1	525.6	103.4	365.9	33532	3645	9.2	538.2	99.4	357.5	0.3	1.1
1212a	135.0	30270	4130	7.3	513.6	105.4	399.0	32169	4072	7.9	541.2	101.1	397.8	6.3	7.8

* 100% (simulated value – tested value)/tested value

Table 4.8 AC model validation with R410 system data
(shaded area indicates tests with a custom-fabricated compressor)

Test number	Outdoor dry-bulb temperature (°F)	Test results						Simulation results						Difference*	
		Capacity (Btu/h)	Work (W)	EER (Btu/h.W)	rmass (lb/h)	Evap. outlet pressure (psi)	Cond. inlet pressure (psi)	Capacity (Btu/h)	Work (W)	EER (Btu/h.W)	rmass (lb/h)	Evap. outlet pressure (psi)	Cond. inlet pressure (psi)	Capacity (%)	EER (%)
b010330k	82.1	40345	2201	18.3	537.0	158.1	341.6	40008	2260	17.7	519.6	153.3	331.8	-0.8	-3.4
b010328a	95.0	38144	2604	14.6	541.8	162.4	399.8	37955	2618	14.5	528.0	157.2	389.0	-0.5	-1.0
b010331x	115.4	33305	3315	10.0	539.4	168.6	507.1	34168	3317	10.3	538.8	162.5	494.0	2.6	2.5
b010403a	125.0	30586	3789	8.1	525.0	170.7	559.1	32260	3708	8.7	543.0	164.7	550.0	5.5	7.8
b010425x	129.9	29414	3963	7.4	521.4	173.0	587.4	31462	3933	8.0	549.0	166.9	581.0	7.0	7.8
c010719a	150.0	24801	5357	4.6	517.2	179.3	709.1	26806	4964	5.4	587.4	171.8	696.1	8.1	16.6
c010723d	155.4	22699	6288	3.6	495.6	182.2	742.8	25253	5261	4.8	610.2	175.7	727.7	11.3	33.3

* 100% (simulated value – tested value)/tested value

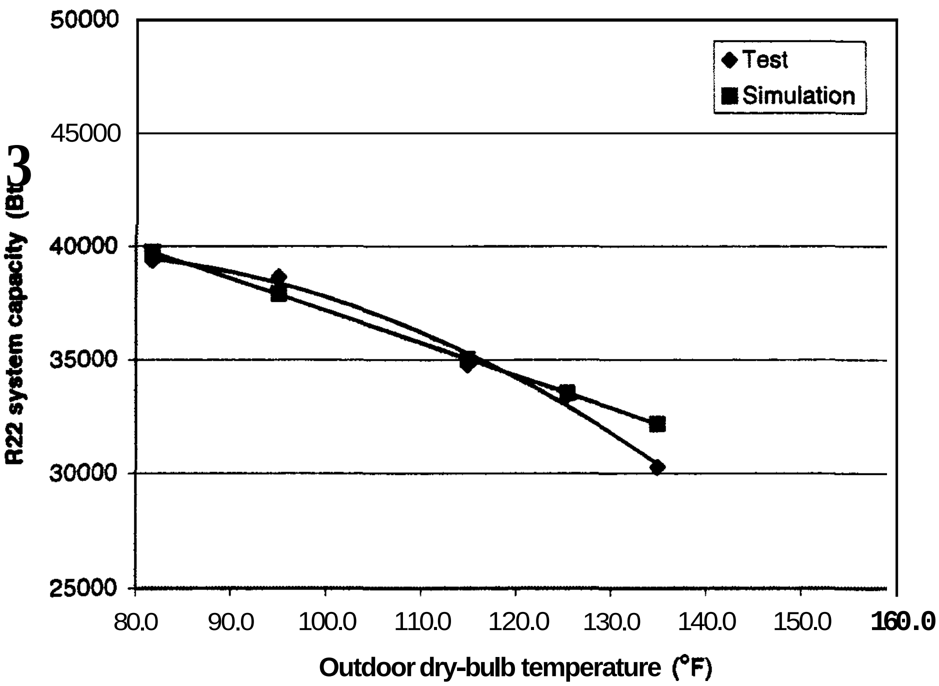


Figure **4.16** Tested and predicted capacities of **R22** air conditioner

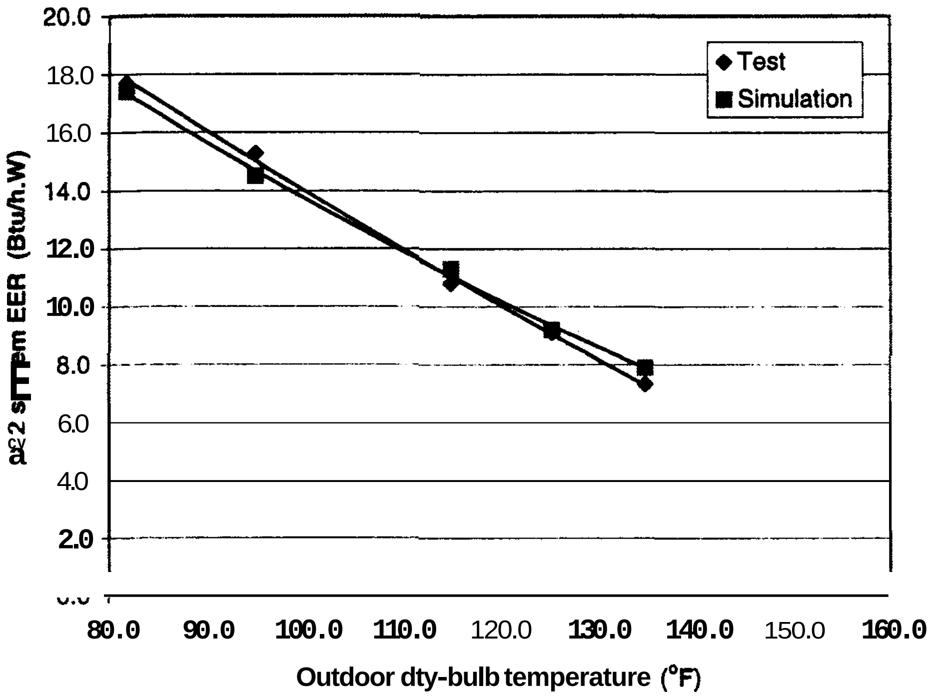


Figure 4.17 Tested and predicted EERs **of R22** air conditioner

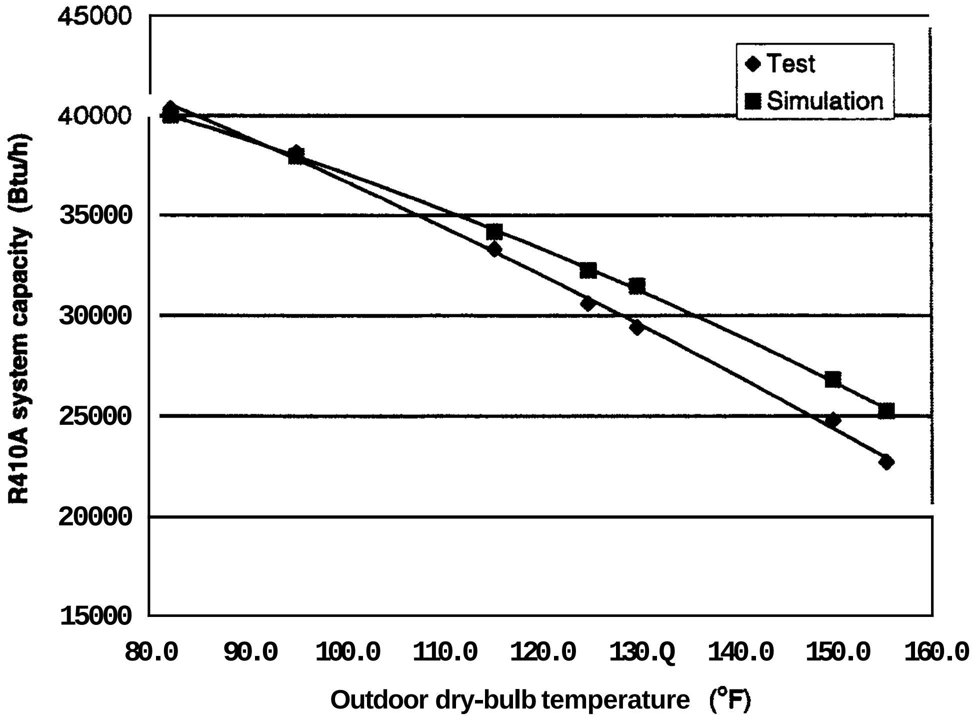


Figure **4.18** Tested and predicted capacities of **R410A** air conditioner

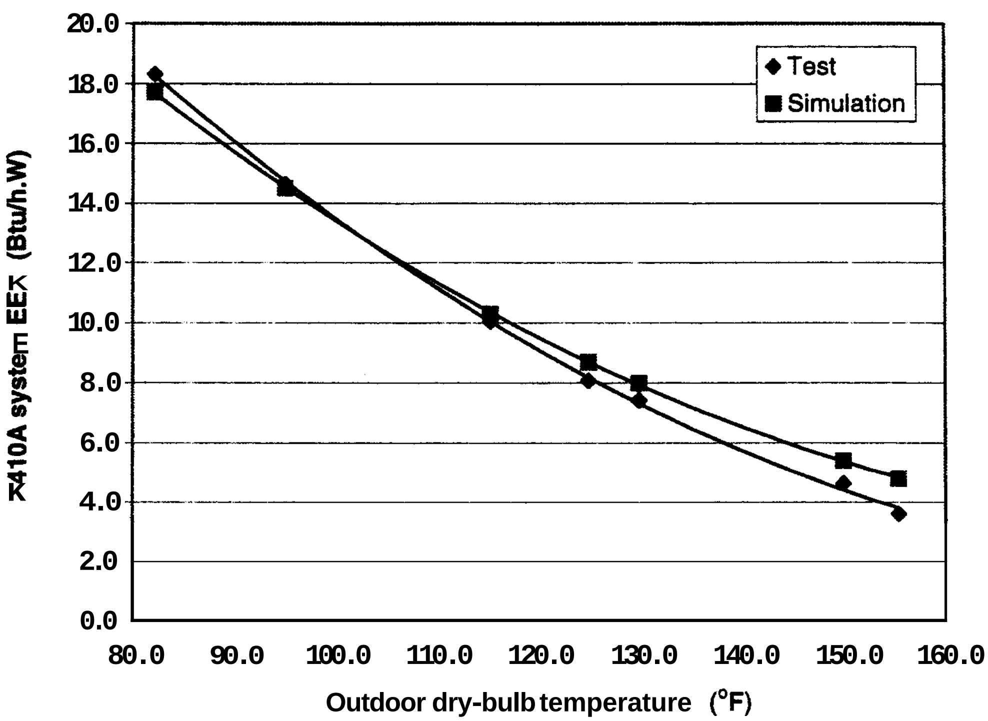


Figure **4.19** Tested and predicted EERs of **R410A** air conditioner

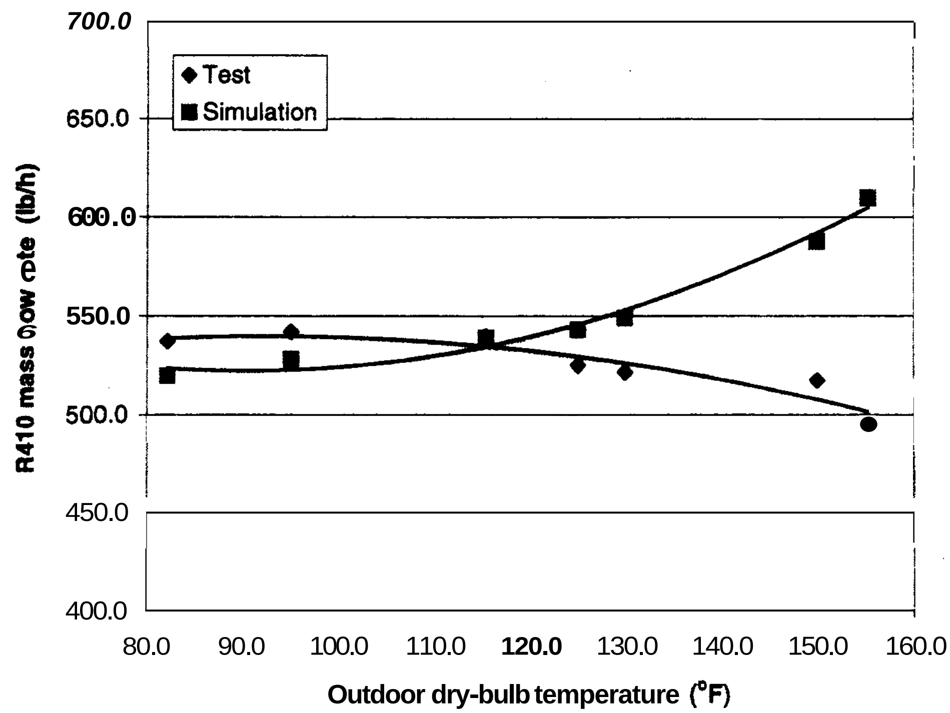


Figure 4.20 Tested and predicted refrigerant mass flow rates for R410A air conditioner

4.5.3 Simulations of R22, R410A, R134a, and R404A systems.

Comparison simulations for R22, R410A, R134a, and R404A systems covered the 82.0 °F to 135.0°F (27.8 °C to 57.2 "C) outdoor temperature range. Each system employed the same heat exchangers as those used in the tested R22 and R410A systems. Refrigerant superheat at the evaporator outlet and subcooling at the condenser outlet were 9 °F (5 "C) for each simulation run. The isentropic efficiency of each compressor was the same as that of the R410A compressor for given suction and discharge saturation temperatures. R22, R404A, and R134a compressors had an adjusted volumetric capacity so each system could deliver the same capacity as the R410A system at the 95.0 "F (35.0 "C) test condition.

Table 4.9 and Figures 4.21 and 4.22 present absolute results of simulations, while Figures 4.23 and 4.24 present capacities and EERs relative to those for the **R22** system. For all systems, capacity displayed a nearly linear dependence on the ambient temperature with R404A having the highest capacity degradation and R22 having the lowest. Regarding efficiency, R410A showed the highest EER at all outdoor temperatures closely followed by R22. These results agree with the findings obtained in the laboratory. R134a was the least efficient fluid at 82.0 °F (27.8 °C) and 95.0 °F (35.0 °C) outdoor temperatures.

The obtained results were affected by a combination of thermophysical refrigerant properties and intangible aspects of system design that would fit these properties better for one refrigerant than the other. We may note that for typical operating conditions, theoretical calculations using thermodynamic properties alone would indicate R134a as the most efficient refrigerant, closely followed by R22, and indicating R410A and R404A as the least efficient refrigerants. However, in the studied system, R134a experienced excessive pressure drop in the heat exchangers, particularly in the evaporator, which resulted in the lowest efficiency. On the other hand it appeared that the circuitry design in the heat exchangers suited well R410A, which had the best overall performance.

Table 4.10 presents selected refrigerant parameters to help to explain the refrigerants' performance in our simulations. For a simplified analysis we may state that a low critical temperature and high molar heat capacity promote high efficiency in a vapor compression cycle. This disadvantages R404A, which has low critical temperature and high molar heat capacity. For R134a, a corollary of its high critical temperature is its low volumetric capacity. Since R134a volumetric flow rate had to be increased to obtain the target capacity of the R410A system at the 95.0 °F (35.0 °C) rating point, excessive pressure drop occurred in the unmodified R134a heat exchangers. These results emphasize the importance of heat exchanger optimization for efficiency improvement of the system.

ulation results for R22, R410A, R 34a. and R404A systems

Refrigerant	Outdoor dry-bulb temperature (°F)	Capacity (Btu/h)	Work (W)	EER (Btu/(Wh))	rmass (lb/h)
R22	82	39295	2299	17.1	523.2
	95	37601	2661	14.1	527.4
	115	34774	3359	10.4	535.8
	125	33470	3805	8.8	540.0
	135	31978	4294	7.4	545.4
R410A	82	39627	2233	17.7	513.0
	95	37582	2595	14.5	521.4
	115	34063	3247	10.5	532.2
	125	32080	3648	8.8	538.2
	135	30040	4103	7.3	543.6
R134a	82	39590	2573	15.4	556.2
	95	37732	2922	12.9	564.0
	115	34485	3638	9.5	576.6
	125	32799	4091	8.0	581.4
	135	31039	4482	6.9	590.4
R404A	82	40486	2496	16.2	721.2
	95	37676	2855	13.2	732.0
	115	33208	3500	9.5	754.2
	125	30714	3895	7.9	768.0
	135	30714	4392	6.4	768.0

Table 4.10 Selected thermodynamic parameters of studied refrigerants

Refrigerant	critical temperature (°F)	Volumetric capacity* (Btu/ft^3)	Molar heat capacity at const. pressure. (kJ/(kmol.K))
R22	205.1	110.2	66.7
R410A	158.3	159.7	85.5
R404A	161.9	112.7	101.1
R134a	213.9	72.0	95.0

*for a basic cycle at 45 °F evaporator sat. temperature and 100 °F condenser sat. temperature, 0 °F evaporator superheat and 0 °F condenser subcooling
** for saturated vapor at 45 °F (280.4 K)

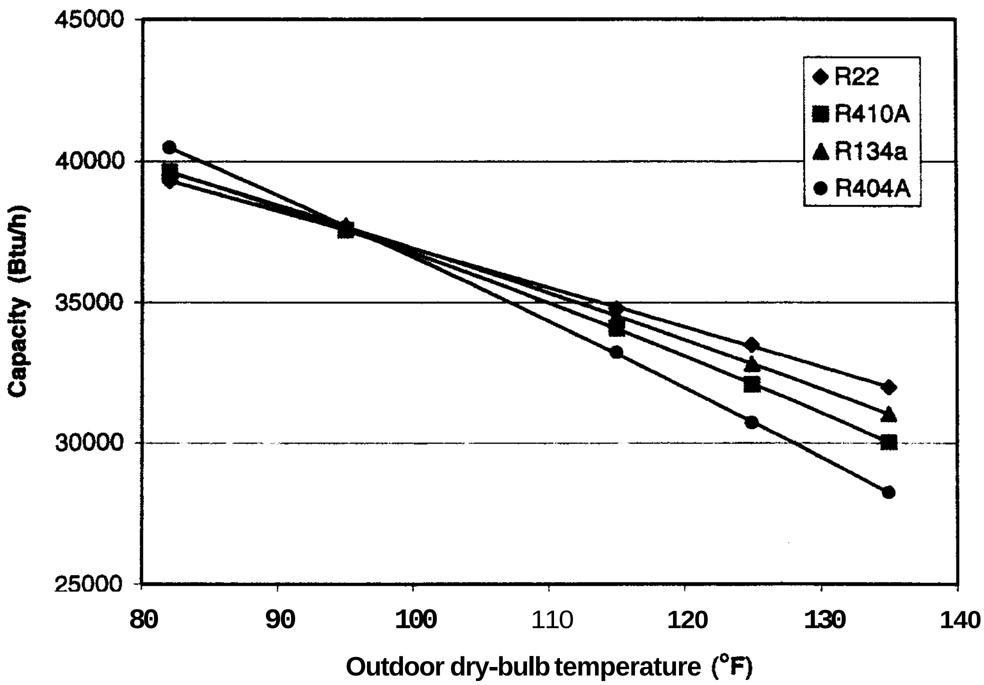


Figure 4.21 Simulated capacities of R22, R410A, R134a, and R404A air conditioners

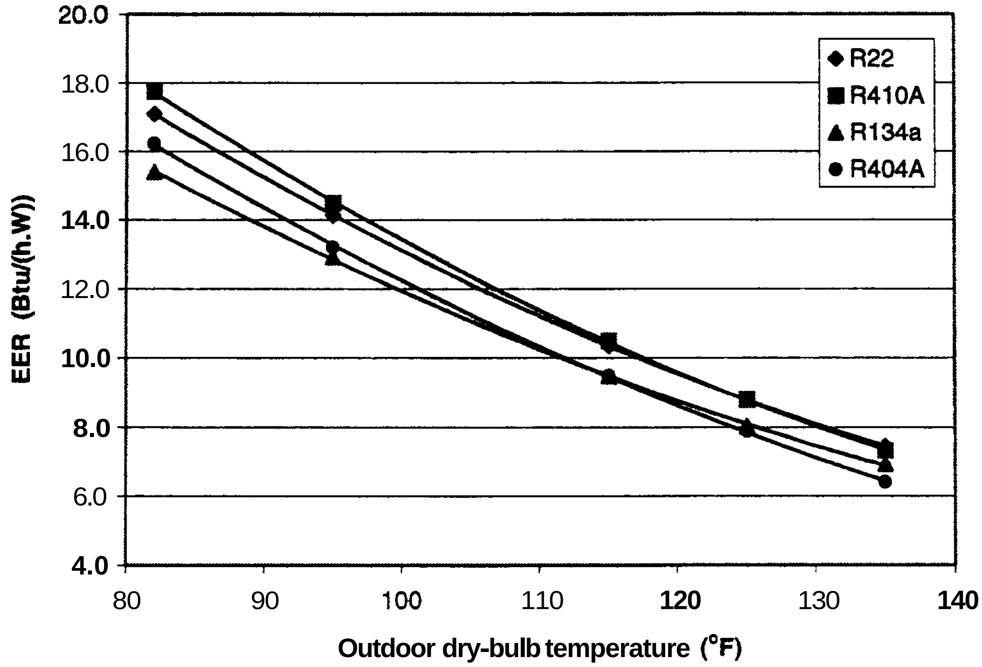


Figure 4.22 Simulated EERs for R22, R410A, R134a, and R404A air conditioners

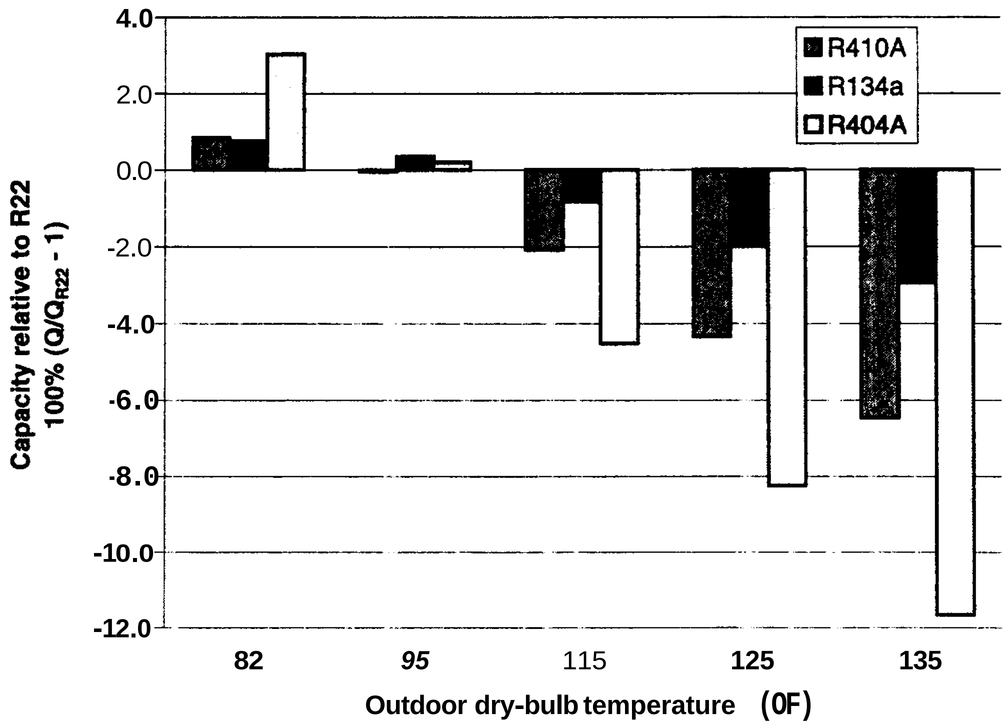


Figure 4.23 Simulated capacities of R410A, R134a, and R404A air conditioners relative to capacity of R22 air conditioner

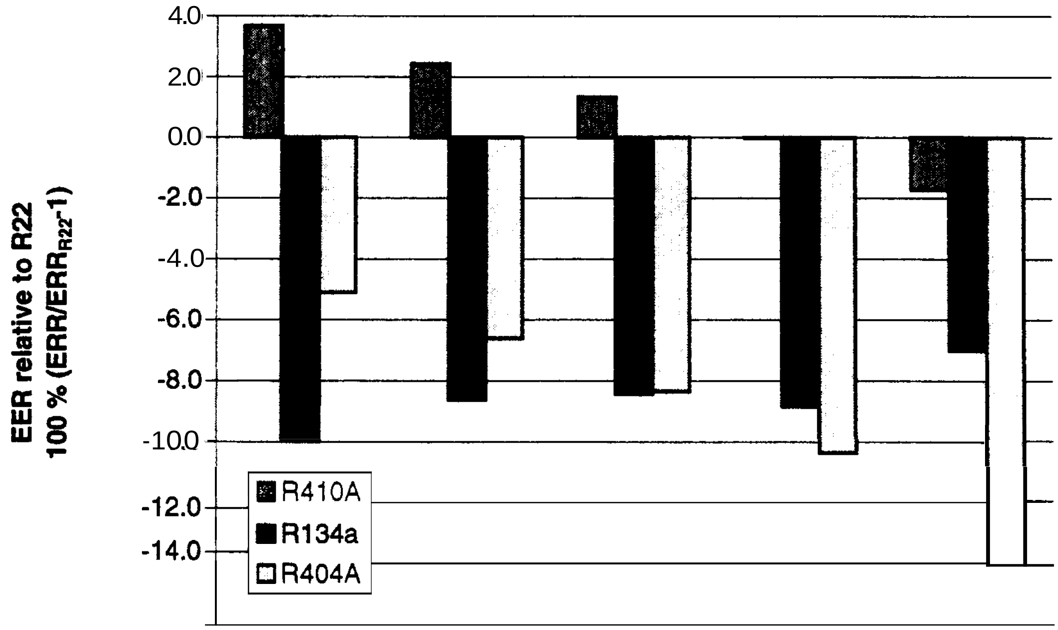


Figure 4.24 Simulated EERs of R410A, R134a, and R404A air conditioners relative to EER of R22 air conditioner

CHAPTER 5. CONCLUSIONS AND SUGGESTIONS FOR FUTURE WORK

5.1 Experimental Work

R22 and R410A split air-conditioning systems were tested and compared as outdoor temperature ranged from 82.0 °F (27.8 °C) to 130 °F (54.4 °C). The R410A system tests were extended to 155.0 °F (68.3 °C) ambient temperature with a customized compressor. When outdoor temperature increased, the R410A system performance degraded more than the R22 system performance. While capacities of both systems were approximately equal at the 95.0 °F (35.0 °C) rating point, at the 130.0 °F (54.4 °C) outdoor temperature the R410A capacity was 9 % below that of R22. For the same test points, the R410A EER (COP) was 4 % and 15 % lower than the R22 EER (COP), respectively. The degradation trend was linear. Since both systems employed identical heat exchangers and similar design (scroll) compressors, the refrigerant and lubricant used in each system had the dominant effect on the measured results. Operation of the R410A system was stable during all tests, including those with the customized compressors extending up to the 155.0 °F (68.3 °C) outdoor temperature and resulting in a supercritical condition at the condenser inlet.

It is evident that the TXV in combination with the large 13 SEER condenser in the current systems were able to maintain subcooled refrigerant at the inlet of the expansion valve. The TXV regulated refrigerant flowrate to prevent flooding the evaporator at the increased ambient temperatures, and the large outdoor condenser was able to subcool the refrigerant. In combination these factors prevented the rapid decline in performance reported by Wells et. al (1999) and predicted by some simulations.

The R410A system with compressor #2 completed tests where the compressor discharge pressure was above the critical point. Tests at an outdoor temperature of 150.0 °F (65.6 "C), 152.0 °F **(66.7 "C),** and 155.0 °F **(68.3 "C)** produced compressor discharge conditions above critical. No noticeable changes in noise level or operation of the system **was** noted. The condenser was able to condense the supercritical vapor and provide subcooling at the TXV inlet. The subcooled liquid at the TXV inlet was the main factor contributing to the stability of the system. A loss of subcooling, due either above critical outlet pressure or incomplete condensation, could have caused mass flow surging and "hunting" of the TXV.

As **an** additional task, it would be of interest to test the same R22 and R410A system with TXVs replaced by short tube restrictors. The fixed area expansion device would not prevent flooding of the evaporator. *Also,* a combination of a smaller area outdoor condensing unit would produce the worst case of two-phase flow in the liquid line. The current evaporator could be fitted with a short tube restrictor and tested with the current R410A condensing unit at the high outdoor temperatures seen during these tests (up to at least 155.0 °F **(68.3 °C)**). Then a smaller outdoor condensing unit could be installed while keeping the same evaporator and short tube restrictor. Modeling could be carried out in parallel to determine how well the current software could be adapted for a fixed area expansion device and near critical conditions.

5.2 Simulation Work

This project provided a thrust for the final stage of preparing a beta version of EVAP-COND, a windows-based simulation package for predicting performance of finned-tube evaporators and condenser. Both the evaporator and condenser models can account for one-dimensional non-uniform air distributions and interaction between the air and refrigerant distributions. The visual interface helps with specifying tube-by-tube refrigerant circuitry and analyzing detailed simulation results on a tube-by-tube basis. Ten refrigerant and refrigerant mixtures are available in EVAP-COND. The package is compatible with REFPROP6 (McLinden et al., 1998), hence any refrigerant and refrigerant mixture covered by the REFPROP6 can included. The condenser and evaporator models were validated with the **R22** and R410A test data showing good and consistent predictions. The validation effort showed the importance of proper representation of the air-side heat transfer and the inadequacy of the currently available correlations.

In the second phase of the modeling effort, we formulated a model for a TXV-equipped air conditioner to simulate system performance for R22, R410A, R404A, and R134a. The model uses the EVAP-COND evaporator and condenser model, and simulates the compressor using a compressor map algorithm. The same as for EVAP-COND, the air conditioner model is REFPROP6-compatible and technically can be used to simulate any refrigerant and refrigerant mixture that is covered by REPFROP6. We validated the system model and performed simulations for the four refrigerants for the 82.0 "F to 135.0 °F (27.8 "C to 57.2 "C) outdoor temperature range using the same heat exchangers as those tested with R22 and R410A. In general, the simulations results are consistent with the test results obtained for R22 and R410A and can be explained in terms of refrigerant thermophysical properties and their impact on performance in a system with non-optimized heat exchangers.

During the development stage of EVAP-COND, we received requests or suggestions for future work from persons who offered to test it. The suggested items for the future work included:

- capability to incorporate new or proprietary air-side and refrigerant side heat transfer correlations by the user of the program
- extension of the evaporator model to the frosting region to allow simulations of evaporators used in heat pumps and commercial refrigeration
- capability to accommodate "non-existing tubes" (empty spaces in the heat exchanger assembly)
- new option for condenser simulation where the outlet subcooling is specified in addition to inlet parameters (The program would iterate the refrigerant mass flow rate that would satisfy the input constraints)
- capability to perform sequential simulation runs.

Additional validations of the evaporator and condenser models would be highly desirable. The additional validations should include different designs, air volumetric flow rates, and tube diameters. We have to recognize that the condenser validation we performed might not be conclusive since the tested condenser had a low approach temperature, and in such situations all simulation models can predict capacity well because of the prediction limit imposed by a pinch point.

The developed simulation model for a TXV-equipped air conditioner can be extended to other expansion devices. Further, the EVAP-COND interface could be utilized as a starting point for a complete window-based heat pump simulation model.

APPENDIX A. SUMMARY OF TEST RESULTS FOR R22 SYSTEM

Summary sheets were generated automatically after each test. For all tests a Coriolis meter was placed in the discharge line to measure mass flowrate in addition to the liquid line Coriolis meter. This redundant measurement was not used for all tests; therefore, the mass flow listed in the summary sheets for the discharge should be ignored.

Table A.1 lists the tests performed with the original R22 compressor and the corresponding outdoor dry-bulb temperatures.

Table A. 1 R22 system tests

Filename	Outdoor Temperature (°F)
A001205a	81.8
A001208a	81.7
A00 1212a	135.0
A001213x	115.0
A001214a	134.8
A00 12 18b	115.1
A010105	95.1
A010108a	115.0
AOlOl 10a	125.4
AOlOl 1l a	81.9
AOlOl 17b	130.5
AOlOl 18x	134.5

```
COOLING TEST SUMMARY SHEET

       DATA FILENAME: a001205a.dat   SUMMARY FILENAME: a001205a.sum
                                                                 Range
 Air-Side Conditions       Range  Total Air-Side Capacity: 38979.04  4568.84
   Indoor Dry-Bulb : 79.625   0.81    Sensible Cap (Btu/h): 28152.04  1258.20
Indoor Inlet Dew (F): 60.477   1.48      Latent Cap (Btu/h): 10827.00  4085.69
Indoor Exit Dry-Bulb: 59.067   0.87     EvapAir Delta T (F):  21.25     0.90
 Indoor Exit Dew (F): 55.444   0.84
 Outdoor Dry-Bulb (F):  81.832  0.57   Sensible Heat Ratio:   0.723    0.0811
 Indoor Airflow (CFM): 1188.35  17.70
 Indoor Airflow (SCFM): 1203.86  16.75      (0.075 lb/ft3 standard air)
   Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011224
    Evap Exit Humidity Ratio (lbH2O/lbAir):   0.009338
    Barometric Pressure (in HG):  29.95      Nozzle Temp (F)   59.74   1 21
       7 inch Nozzle Pressure Drop (in Water):  1.288    0.037
 Evaporator Coil Air Pressure Drop (in Water):  0.182    0.008
--------------------------------------------------------------------------------
 Refrigerant Side Conditions
  Discharge Pressure (psia):  218.82    1.076       Ref-side Cap (Btu/h) : 39975.17   8[illegible].99
          Suction Pressure :   94.78    0.977         Ref-side Cap (tons):     3.33     0.07
   Condenser Inlet Pressure:  218.06    0.977      Liq Line Mdot (lbm/min):    8.93     0.18
    Condenser Exit Pressure:  194.08    1.167     Disch Mass Flow (lb/min):   -0.16     3.84
 Liq MassMeter Inlet (psia):  194.08    1.167    Liq Line Density (lb/ft3):   79.17     0.32
LiqMeter Exit/TXV In (psia):  192.06    0.976    TXV Inlet Temperature (F):   86.87     0.35
                                                  TXV Inlet Subcooling (F):    6.50     0.48
Evaporator Pres Drop (psid):   10.40    1.002  Disch Line Density (lb/ft3):   14.973    0.476
  Evap Exit Pressure (psia):   97.21    1.090             Suction Temp (F):   64.02     1.22
  Evap Exit Temperature (F):   61.69    1.649         Suction Superheat (F):   16.48     1.23
    Evap Exit Superheat (F):   12.65    1.485            Discharge Temp (F):  147.35     0.68
                                                    Discharge Superheat (F):   44.55     0.68
                                                          Cond Inlet Temp (F)  150.16     0.61
                                                           Cond Exit Temp (F)   88.45     0.55
 0.36 WattHours Per Count  ---------------------------------------------------------------------
      Counts :  4264.00                          Test Period (seconds):  2503.78
    WattHours:  1535.04                        Cooling EER:  17.66    COP:   5.18
```

COOLING TEST SUMMARY SHEET

```
        DATA FILENAME: a001208a.dat   SUMMARY FILENAME: a001208a.sum
                                                                       Range
 Air-Side Conditions        Range  Total Air-Side Capacity: 39363.87  1322.19
   Indoor Dry-Bulb : 79.887    0.90    Sensible Cap (Btu/h): 28891.73   857.49
 Indoor Inlet Dew (F)  [illegible]  0 25    Latent Cap (Btu/h): 10472.14   742.52
 Indoor Exit Dry-Bulb  58 [illegible]  [illegible]   EvapAir Delta T (F):  21.87     0.60
 Indoor Exit Dew (F) : 54 [illegible]  0 44
  Outdoor Dry-Bulb (F    81 7[illegible]  0 [illegible]   Sensible Heat Ratio    0 7[illegible]    0 0157
  Indoor Airflow (CFM): 1184.52  1[illegible].29
 Indoor Airflow (SCFM): 1201.08  1[illegible].22      (0 075 lb/f 3 st adard air)
   Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.010992
    Evap Exit Humidity Ratio (lbH2O/lbAir):   0.009164
    Barometric Pressure (in HG)   2[illegible] 95       Nozzle Temp (F):  5[illegible] 3[illegible]   0 [illegible]4
       7 inch Nozzle Pressure Drop (in Water):  1.281     0.035
  Evaporator Coil Air Pressure Drop (in Water):  0.177     0.010
 --------------------------------------------------------------------------------

 Refrigerant Side Conditions
  Discharge Pressure (psia):  217.99    1.907        Ref-side Cap (Btu/h) : [illegible]  102[illegible]
          Suction Pressure :   94.15    0.733          Ref-side Cap (tons):   [illegible]   [illegible]
   Condenser Inlet Pressure:  217.25    1.759       Liq Line Mdot (lbm/min):   8.90      0.23
    Condenser Exit Pressure:  197.52    1.823      Disch Mass Flow (lb/min):  -0.12      0.23
  Liq MassMeter Inlet (psia):  197.52    1.823     Liq Line Density (lb/ft3):  79.12      0.58
 LiqMeter Exit/TXV In (  ia)  189.03    1.904    TXV Inlet Temperature (F):  86.65      0.88
                                                TXV Inlet Subcooling (F):   5.59      0.61
 Evaporator Pres Drop (psid):   10.50    1.072   Disch Line Density (lb/ft3):  18.096    0.183
  Evap Exit Pressure (psia):   96.61    1.090             Suction Temp (F):  63.77      1.73
  Evap Exit Temperature (F):   62.42    1.878         Suction Superheat (F):  16.61      1.85
    Evap Exit Superheat (F):   13.74    2.177           Discharge Temp (F):  [illegible]47.04   1.17
                                                  Discharge Superheat (F):  [illegible]44.52   0.72
                                                      Cond Inlet Temp (F)   [illegible]49.96   0.92
                                                        Cond Exit Temp (F):  88.58      0.86
                                                                         --------------
 0 3[illegible] Watt Hours Per Count  ---
      Counts :  4396.00                         Test Period (seconds):  256[illegible].0[illegible]
     Watt Hours:  1582.56                   Cooling EER:  17.69    COP:   [illegible].1[illegible]
```

COOLING TEST SUMMARY SHEET

```
        DATA FILENAME: a001212a.dat   SUMMARY FILENAME: a001212a.sum
                                                                 Range
  Air-Side Conditions       Range  Total Air-Side Capacity: 30269.60   728.38
    Indoor Dry-Bulb : 79.960   0.24    Sensible Cap (Btu/h): 24807.80   409.92
 Indoor Inlet Dew (F): 60.328   0.20      Latent Cap (Btu/h):  5461.80   693.21
 Indoor Exit Dry-Bulb: 61.944   0.18     EvapAir Delta T (F):  18.79     0.24
  Indoor Exit Dew (F): 57.869   0.21
  Outdoor Dry-Bulb (F): 134.980  1.13   Sensible Heat Ratio    0 820    0 0Z00
  Indoor Airflow (CFM): 1190.48  20.04
 Indoor Airflow (SCFM): 1199.02  19.70      (0.075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011163
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.010208
     Barometric Pressure (in HG):  29.95       Nozzle Temp (F):  ≤2 5Z   0 79
        7 inch Nozzle Pressure Drop (in Water):  1.285    0.043
  Evaporator Coil Air Pressure Drop (in Water):  0.169    0.011
 --------------------------------------------------------------------------------
  Refrigerant Side Conditions
   Discharge Pressure (psia):  399.72    4.254      Ref-side Cap (Btu/h) : 30587.20  12≤3.30
           Suction Pressure :  103.37    0.635        Ref-side Cap (tons):      2.55     0.11
    Condenser Inlet Pressure:  399.04    4.056     Liq Line Mdot (lbm/min):      8.56     0.33
     Condenser Exit Pressure:  378.66    4.618     Disch Mass Flow (lb/min):     0.22     3.88
  Liq MassMeter Inlet (psia):  378.66    4.618    Liq Line Density (lb/ft3):   105.28     0.76
 LiqMeter Exit/TXV In (psia):  375.21    5.370    TXV Inlet Temperature (F):   134.18     0.56
                                                   TXV Inlet Subcooling (F):    11.16     1.03
 Evaporator Pres Drop (psid):   17.99    1 825  Disch Line Density (lb/ft3):     5.145    0.985
   Evap Exit Pressure (psia):  105.35    1 187            Suction Temp (F):     77.49     2.16
   Evap Exit Temperature (F):   65.48    3 368       Suction Superheat (F):     24.75     1.93
     Evap Exit Superheat (F):   11.60    3 103          Discharge Temp (F):    232.32     1.44
                                                    Discharge Superheat (F):    81.60     1.46
                                                        Cond Inlet Temp (F):   237.17     1.25
                                                         Cond Exit Temp (F):   141.67     0.95
  0 36 WattHours Per Count  -------------------------------------------------------------------
        Counts :  7255.00                           Test Period (seconds):  2277.54
      WattHours:  2611.80                          Cooling EER:   7.33    COP:   2.15
```

COOLING TEST SUMMARY SHEET

DATA FILENAME: a001213x.dat   SUMMARY FILENAME: a001213x.sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb :	80.240	0.91	Total Air-Side Capacity:	34264.43	2651.27
Indoor Inlet Dew (F)	59 336	0 54	Sensible Cap (Btu/h):	27492.35	1093.68
Indoor Exit Dry-Bulb	60 392	0 45	Latent Cap (Btu/h):	6772.08	1990.76
Indoor Exit Dew (F):	56.187	0.55	EvapAir Delta T (F):	20.61	0.78
Outdoor Dry-Bulb (F):	114.963	0.91	Sensible Heat Ratio	0 303	0 0459
Indoor Airflow (CFM):	1199.64	12.12			
Indoor Airflow (SCFM	1212 26	12 77	(0.075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir): 0.010768

Evap Exit Humidity Ratio (lbH2O/lbAir): 0.009597

Barometric Pressure (in HG): 29.95   Nozzle Temp (F): 50 97   0 90

7 inch Nozzle Pressure Drop (in Water): 1.309   0.027

Evaporator Coil Air Pressure Drop (in Water): 0.170   0.007

Refrigerant Side Conditions					
Discharge Pressure (psia):	319.81	2.054	Ref-side Cap (Btu/h) :	34338.16	1632.22
Suction Pressure :	99.41	0.635	Ref-side Cap (tons):	2.86	0.14
Condenser Inlet Pressure:	319.09	1.955	Liq Line Mdot (lbm/min):	8.81	0.40
Condenser Exit Pressure:	300.31	1.872	Disch Mass Flow (lb/min):	-0.07	1.27
Liq MassMeter Inlet (psia):	300.31	1.872	Liq Line Density (lb/ft3):	90.54	1.64
LiqMeter Exit/TXV In (psia):	291.22	2.099	TXV Inlet Temperature (F):	118.32	0.68
			TXV Inlet Subcooling (F):	6.30	0.46
Evaporator Pres Drop (psid):	16.87	1.959	Disch Line Density (lb/ft3):	29.904	0.000
Evap Exit Pressure (psia):	101.74	0.605	Suction Temp (F):	72.29	1.24
Evap Exit Temperature (F):	67.16	1.126	Suction Superheat (F):	21.91	1.45
Evap Exit Superheat (F):	15.38	1.026	Discharge Temp (F):	193.37	0.88
			Discharge Superheat (F):	61.25	0.81
			Cond Inlet Temp (F)	197.24	0.69
			Cond Exit Temp (F):	123.89	0.45

0 36 WattHours Per Count

Counts : 3209.00

WattHours: 1135.24

Test Period (seconds): 1326.62

Cooling EER: 10.93   COP: 3.20

COOLING TEST SUMMARY SHEET

DATA FILENAME: a0 1[illegible]1 a.dat   SUMMARY FILENAME a001[illegible]14a sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb :	80.465	0.30	Total Air-Side Capacity:	29711.40	635.35
Indoor Inlet Dew (F	[illegible]9 9[illegible]1	0 [illegible]0	Sensible Cap (Btu/h):	25364.09	303.65
Indoor Exit Dry-Bulb:	62.119	0.17	Latent Cap (Btu/h):	4347.32	529.95
Indoor Exit Dew (F):	57.962	0.15	EvapAir Delta T (F):	19.14	0.14
Outdoor Dry-Bulb (F):	134.830	0.[illegible]5	Sensible Heat Ratio:	0 854	0 0151
Indoor Airflow (CFM):	1195.23	12.84			
Indoor Airflow (SCFM	1[illegible]0[illegible] [illegible]6	1[illegible] 67	(0.075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir):	0.011000	
Evap Exit Humidity Ratio (lbH2O/lbAir):	0.010243	
Barometric Pressure (in HG): 29.95 Nozzle Temp (F):	[illegible] 57	0 5[illegible]
7 inch Nozzle Pressure Drop (in Water):	1.295	0 027
Evaporator Coil Air Pressure Drop (in Water):	0.169	0.01[illegible]

Refrigerant Side Conditions					
Discharge Pressure (psia):	396.83	1.956	Ref-side Cap (Btu/h)	30405.03	1538.98
Suction Pressure :	104.21	0.635	Ref-side Cap (tons):	2.53	0.13
Condenser Inlet Pressure:	396.13	1.955	Liq Line Mdot (lbm/min):	8.61	0.43
Condenser Exit Pressure	376.56	1.945	Disch Mass Flow (lb/min):	-0.01	1.72
Liq MassMeter Inlet (psia):	376.56	1.945	Liq Line Density (lb/ft3):	[illegible]01.21	1.70
LiqMeter Exit/TXV In (psia):	369.77	1.953	TXV Inlet Temperature (F):	[illegible]36.93	0.55
			TXV Inlet Subcooling (F):	7.18	0.65
Evaporator Pres Drop (psid):	19.30	2.125	Disch Line Density (lb/ft3):	29.904	0.000
Evap Exit Pressure (psia):	106.31	0.848	Suction Temp (F):	78.44	1.13
Evap Exit Temperature (F):	68.76	0.614	Suction Superheat (F):	25.22	1.16
Evap Exit Superheat (F):	14.33	0.883	Discharge Temp (F):	[illegible]30.06	0.70
			Discharge Superheat (F):	79.96	0.74
			Cond Inlet Temp (F):	235.01	0.60
			Cond Exit Temp (F):	143.64	0.53

0.36 WattHours Per Count
Counts : 6011.00
WattHours: 2163.96

Test Period (seconds): 1906.41
Cooling EER: 7.27 COP: 2.13

```
COOLING TEST          SYSTEM

        DATA FILENAME: a001218b.dat   SUMMARY FILENAME: a001218b.sum
                                                                       Range
  Air-Side Conditions        Range  Total Air-Side Capacit : 33843.68  2537.11
    Indoor Dry-Bulb : 80.063   2.75    Sensible Cap (Btu/h : 26957.45  2696.57
 Indoor Inlet Dew (F): 59.655   0.20      Latent Cap  Btu/h :  6886.23  1402.67
 Indoor Exit Dry-Bulb: 60.444   0.73     EvapAir Delta T (F :  20.3      2.01
  Indoor Exit Dew (F): 56.457   0.60
   Outdoor Dry-Bulb (F): 115.057  0.69   Sensible Heat Ratio    0 797    0 0445
   Indoor Airflow (CFM): 1190.45  10.25
  Indoor Airflow (SCFM): 1202.40  10.41      (0 075 lb/ft3 standard air)
     Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.010894
      Evap Exit Humidity Ratio (lbH2O/lbAir):   0.009693
      Barometric Pressure (in HG):  29.95       Nozzle Temp (F):  61 19   1 13
         7 inch Nozzle Pressure Drop (in Water):  1.289     0.022
   Evaporator Coil Air Pressure Drop (in Water):  0.171     0.010
                                                                  ------------------

   Refrigerant Side Conditions
    Discharge Pressure (psia):  321.20     1.956       Ref-side Cap (Btu/h) : 34130.11  1319.37
            Suction Pressure :   99.49     1.222         Ref-side Cap (tons):      2.84      0.11
     Condenser Inlet Pressure:  320.43     1.955      Liq Line Mdot (lbm/min):      8.76      0.32
      Condenser Exit Pressure:  301.46     1.823     Disch Mass Flow (lb/min):     -0.11      1.24
   Liq MassMeter Inlet (psia):  301.46     1.823    Liq Line Density (lb/ft3):     89.68      1.84
  LiqMeter Exit/TXV In (psia):  291.00     2.441    TXV Inlet Temperature (F):    118.42      0.60
                                                     TXV Inlet Subcooling (F):      6.14      0.68
  Evaporator Pres Drop (psid):   17.02     1.2?5   Disch Line Density (lb/ft3):    29.904     0.000
    Evap Exit Pressure (psia):  101.85     1.4?3             Suction Temp (F):     72.40      2.17
    Evap Exit Temperature (F):   66.82     2.3?7         Suction Superheat (F):     21.97      1.53
      Evap Exit Superheat (F):   14.98     1.9?7            Discharge Temp (F):    193.96      0.85
                                                        Discharge Superheat (F):     61.48      0.68
                                                            Cond Inlet Temp (F):    197.91      0.95
                                                             Cond Exit Temp (F):    124.02      0.68
   0 36 WattHours Per Count  ------------                ---------------------------------------
        Counts :  3560.00                              Test Period (seconds):  14?3.61
     WattHours:  1281.60                            Cooling EER:  10.7?     COP     3.14
```

COOLING TEST SUMMARY SHEET

DATA FILENAME: a010105.dat   SUMMARY FILENAME: a010105.sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb	79 [illegible]3	0.83	Total Air-Side Capacity:	38639.91	1209[illegible]01
Indoor Inlet [illegible] ([illegible].	53 951	0.74	Sensible Cap (Btu/h):	28306.62	1010.49
Indoor Exit Dry-Bulb:	58.961	0.65	Latent Cap (Btu/h):	10333.29	1379.93
Indoor Exit Dew ([illegible]	35 02[illegible]	0.80	EvapAir Delta T (F):	21.57	0.80
Outdoor Dry-Bulb ([illegible]	95 051	1.73	Sensible Heat Ratio	0 733	0 0 1[illegible]
Indoor Airflow (CFM):	1177.33	14.49			
Indoor Airflow (SCFM)	1192 64	14.91	(0 075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir):	0.011012	
Evap Exit Humidity Ratio (lbH2O/lbAir):	0.009195	
Barometric Pressure (in HG): 29.95	Nozzle Temp (F): [illegible]9 81	0 92
7 inch Nozzle Pressure Drop (in Water):	1.264	0.031
Evaporator Coil Air Pressure Drop (in Water):	0.176	0.010

Refrigerant Side Conditions

Discharge Pressure (psia):	259.75	3.831	Ref-side Cap (Btu/h):	38209.50	1706.83
Suction Pressure :	96.35	1.954	Ref-side Cap (tons):	3.19	0.14
Condenser Inlet Pressure:	259.09	3.746	Liq Line Mdot (lbm/min):	8.96	0.37
Condenser Exit Pressure:	240.50	4.132	Disch Mass Flow (lb/min):	-0.06	2.01
Liq MassMeter Inlet (psia):	240.50	4.132	Liq Line Density (lb/ft3):	84.68	1.00
LiqMeter Exit/TXV In (psia):	233.08	3.905	TXV Inlet Temperature (F):	97.75	1.43
			TXV Inlet Subcooling (F):	9.73	1.44
Evaporator Pres Drop (psid):	12.65	1.290	Disch Line Density (lb/ft3):	29.904	0.000
Evap Exit Pressure (psia):	98.81	2.260	Suction Temp (F):	65.65	2.64
Evap Exit Temperature (F):	61.53	3.388	Suction Superheat (F):	17.13	2.51
Evap Exit Superheat (F):	11.50	3.374	Discharge Temp (F):	165.28	1.31
			Discharge Superheat (F):	49.59	1.47
			Cond Inlet Temp (F):	168.39	1.49
			Cond Exit Temp (F):	99.97	1.40

0 3[illegible] Watt Hours Per Count

Counts : 5961.00

WattHours: 2145.96

Test Period (seconds): 3055.72

Cooling EER : 15.28   COP: 4.48

COOLING TEST SUMMARY SHEET

DATA FILENAME a010108a.dat   SUMMARY FILENAME a010103a sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb :	79.706	1.02	Total Air-Side Capacity:	34781.93	1626.60
Indoor Inlet Dew (F)	59.872	1.13	Sensible Cap (Btu/h):	26783.85	1228.79
Indoor Exit Dry-Bulb	60.082	1.12	Latent Cap (Btu/h):	7998.08	1887.90
Indoor Exit Dew (F)	56.121	1.23	EvapAir Delta T (F):	20.41	0 81
Outdoor Dry-Bulb (F):	115.097	0.71	Sensible Heat Ratio:	0 770	0 0430
Indoor Airflow (CFM):	1179.84	19.58			
Indoor Airflow (SCFM):	1192.62	20.74	(0.075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir): 0.010981
Evap Exit Humidity Ratio (lbH2O/lbAir): 0.009575
Barometric Pressure (in HG): 29.95   Nozzle Temp (F): [illegible]0.8Z 1 [illegible]
7 inch Nozzle Pressure Drop (in Water): 1.267 0.043
Evaporator Coil Air Pressure Drop (in Water): 0.172 0.01[illegible]

Refrigerant Side Conditions					
Discharge Pressure (psia):	327.01	2.038	Ref-side Cap (Btu/h):	30918.81	2703.31
Suction Pressure :	99.23	1.954	Ref-side Cap (tons):	2.91	0.23
Condenser Inlet Pressure:	326.36	2.199	Liq Line Mdot (lbm/min):	8.83	0.70
Condenser Exit Pressure:	307.44	2.066	Disch Mass Flow (lb/min):	-0.05	1.35
Liq MassMeter Inlet (psia):	307.44	2.066	Liq Line Density (lb/ft3):	94.16	0.52
LiqMeter Exit/TXV In (psia):	301.76	2.929	TXV Inlet Temperature (F):	114.47	0.72
			TXV Inlet Subcooling (F):	12.98	1.02
Evaporator Pres Drop (psid):	15.60	2.064	Disch Line Density (lb/ft3):	29.904	0.000
Evap Exit Pressure (psia):	101.42	2.583	Suction Temp (F):	70.77	2.63
Evap Exit Temperature (F):	61.85	4.308	Suction Superheat (F):	20.50	2.22
Evap Exit Superheat (F):	10.26	3.719	Discharge Temp (F):	196.02	1.61
			Discharge Superheat (F):	62.08	1.83
			Cond Inlet Temp (F):	199.84	1.58
			Cond Exit Temp (F):	119.64	0.80

0 3[illegible] WattHours Per Count ---

Counts : 14092.00   Test Period (seconds): 5670.35

WattHours: 5073 12   Cooling EER: 10.80 COP 3.16

COOLING TEST SUMMARY SHEET

```
        DATA FILENAME: a010110a.dat   SUMMARY FILENAME: a010110a.sum
                                                                         Range
  Air-Side Conditions        Range  Total Air-Side Capacity: 33420.75  1208.28
    Indoor Dry-Bulb : 79.728   1.13    Sensible Cap (Btu/h): 25658.78   861.22
Indoor Inlet Dew (F): 60.495   0.30      Latent Cap (Btu/h):  7761.97   931.44
Indoor Exit Dry-Bulb: 61.086   0.36     EvapAir Delta T (F):  19.39     0.73
 Indoor Exit Dew (F)  56 9≤9   0 ∃0
  Outdoor Dry-Bulb  F): 125 412  0 ≒∃   Sensible Heat Ratio    0 7≤8    0 0202
  Indoor Airflow (CFM): 1191.16  16.86
 Indoor Airflow (SCFM)  1201 77  15 54      (0 075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011231
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.009877
     Barometric Pressure (in HG):  29.95       Nozzle Temp (F):  61 7Z   0 98
        7 inch Nozzle Pressure Drop (in Water):  1.289    0.035
  Evaporator Coil Air Pressure Drop (in Water):  0.172    0.012

  Refrigerant Side Conditions
   Discharge Pressure (psia):  366.58    2.445        Ref-side Cap (Btu/h) : 33143.49  1851.82
           Suction Pressure :  101.49    0.977          Ref-side Cap (tons):     2.76     0.15
    Condenser Inlet Pressure:  365.91    1.955      Liq Line Mdot (lbm/min):     8.76     0.49
     Condenser Exit Pressure:  346.87    2.552     Disch Mass Flow (lb/min):    -0.07     0.47
  Liq MassMeter Inlet (psia):  346.87    2.552    Liq Line Density (lb/ft3):    99.12     0.48
 LiqMeter Exit/TXV In (psia):  342.02    3.417    TXV Inlet Temperature (F):   123.26     0.54
                                                   TXV Inlet Subcooling (F):    14.36     1.21
 Evaporator Pres Drop (psid):   16.84    2.102   Disch Line Density (lb/ft3):   29.904    0.000
   Evap Exit Pressure (psia):  103.42    1.453              Suction Temp (F):    74.17     2.68
   Evap Exit Temperature (F):   62.52    3.843         Suction Superheat (F):    22.54     2.39
     Evap Exit Superheat (F):    9.76    3.891            Discharge Temp (F):   214.21     1.48
                                                     Discharge Superheat (F):    70.82     1.37
                                                         Cond Inlet Temp (F):   218.59     1.06
                                                          Cond Exit Temp (F):   129.91     0.51
  0 36 WattHours Per Count  ----                                            -------------------
        Counts :  9124.00                        Test Period (seconds):  3218.75
      WattHours:  3284.64                      Cooling EER:   9.10    COP:   2.67
```

COOLI TEST SUMMARY SHEET

DATA FILENAME: a010111a.dat   SUMMARY FILENAME: a010111a.sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb	80 311	1 04	Total Air-Side Capacity:	40200.67	1282.92
Indoor Inlet Dew (F)	50 08	0 30	Sensible Cap (Btu/h):	29184.32	1080.80
Indoor Exit Dry-Bulb:	53.036	0.41	Latent Cap (Btu/h):	11016.35	818.20
Indoor Exit Dew (F):	50.888	0.26	EvapAir Delta T (F):	22.02	0.78
Outdoor Dry-Bulb (F)	81 835	0 71	Sensible Heat Ratio	0 72	0 0132
Indoor Airflow (CFM	1188 87	22 77			
Indoor Airflow (SCFM):	1204.59	23.06	(0 075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir): 0.011066

Evap Exit Humidity Ratio (lbH2O/lbAir): 0.009148

Barometric Pressure (in HG): 29.95   Nozzle Temp (F): 39.71   0 30

7 inch Nozzle Pressure Drop (in Water): 1.289   0.049

Evaporator Coil Air Pressure Drop (in Water): 0.179   0.013

Refrigerant Side Conditions

Discharge Pressure (psia):	220.16	1.793	Ref-side Cap (Btu/h):	40405.13	1817.44
Suction Pressure:	94.94	0.977	Ref-side Cap (tons):	3.37	0.15
Condenser Inlet Pressure:	220.46	1.629	Liq Line Mdot (lbm/min):	8.95	0.37
Condenser Exit Pressure:	203.04	1.580	Disch Mass Flow (lb/min):	0.04	0.77
Liq MassMeter Inlet (psia):	203.04	1.580	Liq Line Density (lb/ft3):	78.31	0.44
LiqMeter Exit/TXV In (psia):	193.33	2.115	TXV Inlet Temperature (F):	85.42	0.59
			TXV Inlet Subcooling (F):	8.43	0.72
Evaporator Pres Drop (psid):	10.48	1.581	Disch Line Density (lb/ft3):	29.904	0.000
Evap Exit Pressure (psia):	97.27	1.574	Suction Temp (F):	63.68	2.82
Evap Exit Temperature (F):	65.41	3.550	Suction Superheat (F):	16.04	2.84
Evap Exit Superheat (F):	16.32	3.417	Discharge Temp (F):	147.99	0.81
			Discharge Superheat (F):	44.73	1.10
			Cond Inlet Temp (F):	150.64	1.00
			Cond Exit Temp (F)	86.89	0.77

0.36 WattHours Per Count -------

Counts : 6878.00

WattHours: 2476.08

Test Period (seconds): 4050.54

Cooling EER: 18.29   COP: 3.36

COOLING TEST SUMMARY SHEET

DATA FILENAME a010117b dat SUMMARY FILENAME a010117b sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb	80 101	0.87	Total Air-Side Capacity:	31607.10	1257.21
Indoor Inlet Dew (F)	5a 882	0 30	Sensible Cap (Btu/h):	25718.85	989.60
Indoor Exit Dry-Bulb:	51 442	0 50	Latent Cap (Btu/h):	5888.24	887.11
Indoor Exit Dew (F)	57 181	0 39	EvapAir Delta T (F)	19.42	0.57
Outdoor Dry-Bulb (F)	130 483	0 48	Sensible Heat Ratio	0 814	0 0228
Indoor Airflow (CFM):	1193.27	15.39			
Indoor Airflow (SCFM)	1202 90	16 59	(0 075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir): 0.010984

Evap Exit Humidity Ratio (lbH2O/lbAir): 0.009958

Barometric Pressure (in HG) 29 35 Nozzle Temp (F): 52 13 1 10

7 inch Nozzle Pressure Drop (in Water): 1.292 0 134

Evaporator Coil Air Pressure Drop (in Water): 0.166 0.110

Refrigerant Side Conditions

Discharge Pressure (psia):	385.78	1.712	Ref-side Cap (Btu/h):	31732.29	506.38
Suction Pressure:	101.94	1.140	Ref-side Cap (tons):	2.65	0.13
Condenser Inlet Pressure:	385.14	1.547	Liq Line Mdot (lbm/min):	8.61	0.44
Condenser Exit Pressure:	366.17	2.188	Disch Mass Flow (lb/min):	-0.03	1.17
Liq MassMeter Inlet (psia):	366.17	2.188	Liq Line Density (lb/ft3):	101.86	0.25
LiqMeter Exit/TXV In (psia):	362.20	2.685	TXV Inlet Temperature (F):	127.77	0.57
			TXV Inlet Subcooling (F):	14.60	0.96
Evaporator Pres Drop (psid):	17.35	2.393	Disch Line Density (lb/ft3):	29.904	0.000
Evap Exit Pressure (psia):	103.99	1.574	Suction Temp (F):	75.61	1.80
Evap Exit Temperature (F):	62.29	3.660	Suction Superheat (F):	23.72	1.66
Evap Exit Superheat (F):	9.19	3.526	Discharge Temp (F):	224.42	0.93
			Discharge Superheat (F):	76.72	1.01
			Comp Inlet Temp (F):	229.39	0.93
			Comp Exit Temp (F):	135.20	0.51

0 35 WattHours Per Count ----

Counts : 3921.00

WattHours: 1411.55

Test Period (seconds): 1254.50

Cooling E : 7.80 COP: 2.29

```
COOLING TEST  SUMMARY SHEET

        DATA FILENAME: a010118x.dat   SUMMARY FILENAME: a010118x.sum
                                                               Range
 Air-Side Conditions        Range  Total Air-Side Capacity: 30668.59   744.39
   Indoor Dry-Bulb : 79.793   0.85    Sensible Cap (Btu/h): 25169.33   730.27
Indoor Inlet Dew (F): 59.886  0.05      Latent Cap (Btu/h):  5499.26   509.27
Indoor Exit Dry-Bulb: 61.618  0.35     EvapAir Delta T (F):  19.00     0.64
 Indoor Exit Dew (F): 57.382  0.29
  Outdoor Dry-Bulb (F): 134.487  0.73   Sensible Heat Ratio:   0 82    0 014
  Indoor Airflow (CFM): 1194.35  10.50
 Indoor Airflow (SCFM): 1203.42   9.93     (0.075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.010986
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.010028
     Barometric Pressure (in HG):  29.95      Nozzle Temp (F):  62 35   0 4
        7 inch Nozzle Pressure Drop (in Water):  1.294     0.022
  Evaporator Coil Air Pressure Drop (in Water):  0.169     0.006
 -------------------------------------         ------------------

  Refrigerant Side Conditions
   Discharge Pressure (psia):  401.94     2.771       Ref-side Cap (Btu/h) : 31071.02  1867.35
           Suction Pressure :  102.82     0.977        Ref-side Cap (tons):     2.59     0.16
    Condenser Inlet Pressure:  401.22     2.932     Liq Line Mdot (lbm/min):    8.54     0.49
     Condenser Exit Pressure:  382.05     3.119     Disch Mass Flow (lb/min):  -0.02     0.99
  Liq MassMeter Inlet (psia):  382.05     3.119    Liq Line Density (lb/ft3):  103.77    0.28
 LiqMeter Exit/TXV In (psia):  378.55     4.393    TXV Inlet Temperature (F):  131.13    0.42
                                                    TXV Inlet Subcooling (F):  14.96     0.76
 Evaporator Pres Drop (psid):   17.66     1.961  Disch Line Density (lb/ft3):  29.904    0.000
   Evap Exit Pressure (psia):  104.61     1.574            Suction Temp (F):   76.59     2.34
   Evap Exit Temperature (F):   65.54     2.734        Suction Superheat (F):  24.18     1.91
     Evap Exit Superheat (F):   12.08     2.785           Discharge Temp (F):  232.52    1.46
                                                     Discharge Superheat (F):  81.32     1.22
                                                          Cond Inlet Temp (F):  237.62    1.30
                                                           Cond Exit Temp (F):  138.51    0.90

  0.36 WattHours Per Count
        Counts :  4679.00                        Test Period (seconds):  1467.89
      WattHours:  1684.44                      Cooling EER:    7.42    COP:   2.18
```

APPENDIX B. SUMMARY OF TEST RESULTS FOR R410A SYSTEM

Summary sheets were generated automatically after each test. For all tests a Coriolis meter was placed in the discharge line to measure mass flowrate in addition to the liquid line Coriolis meter. This redundant measurement was not used for all tests; therefore, the mass flow listed in the summary sheets for the discharge should be ignored.

B.1 R410A System With Original Compressor

Table B.1 lists the tests performed with the original R410A compressor and the corresponding outdoor dry-bulb temperatures.

Table B.1 R410A tests with compressor #1

Filename	Outdoor Temperature (“F)
B010320a	95.1
BO10328a	95.0
B010329x	125.5
BO10329b	125.4
B010330k	82.1
B010331x	115.4
B010402a	129.8
BO10403a	125.0
B010410a	114.9
B010425x	129.9

```
COOLING TEST SUMMARY SHEET

        DATA FILENAME  b010320a.dat    t[redacted]   FILENAME  b010320a.sum
                                                                  Range
 Air-Side Conditions        Range  Total Air-Side Capacity: 37254.43   856.57
   Indoor Dry-Bulb : 79.748   1.04    Sensible Cap (Btu/h): 28509.43  1074.69
Indoor Inlet Dew (F): 59.725   0.10      Latent Cap (Btu/h):  8745.00   662.27
Indoor Exit Dry-Bulb: 58.831   0.33     EvapAir Delta T (F):  21.67     0.76
 I door Exit Dew (F   55 583   0 30
  Outdoor Dry-Bulb (F)   95 141   1 03   Sensible Heat Ratio:   0 765    0 0165
  Indoor Airflow (CFM)  1173 62   12 00
 Indoor Airflow (SCFM)  1135 68   11 32      ( 0 075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.010922
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.009388
     Barometric Pressure (in HG :  23 35       Nozzle Temp (F):  59 48   0.66
        7 inch Nozzle Pressure Drop (in Water):  1.270     0.025
  Evaporator Coil Air Pressure Drop (in Water):  0.148     0.009
  -----------------------------------------         ------

 Refrigerant Side Conditions
   Discharge Pressure (psia):  398.98    4.159         Ref-side Cap (Btu/h)   37765 34    527 16
           Suction Pressure :  159.48    0.629          Ref-side Cap (tons):      3.15      0.04
    Condenser Inlet Pressure:  398.25    3.964       Liq Line Mdot (lbm/min):     8.97      0.12
     Condenser Exit Pressure:  386.87    4.055      Disch Mass Flow (lb/min):    -0.10      0.00
  Liq MassMeter Inlet (psia):  386.87    4.055     Liq Line Density (lb/ft3):    76.57      0.23
 LiqMeter Exit/TXV In (psia):  373.89    4.260     TXV Inlet Temperature (F):    99.54      0.51
                                                     TXV Inlet Subcooling (F):     9.27      1.28
 Evaporator Pres Drop (psid):    4.68    0.359   Disch Line Density (lb/ft3):   19.377     0.377
   Evap Exit Pressure (psia):  161.11    1.095             Suction Temp (F):    56.91      2.30
   Evap Exit Temperature (F):   55.37    4.708         Suction Superheat (F):     5.94      2.15
     Evap Exit Superheat (F):    3.77    4.940           Discharge Temp (F):   158.87      1.11
                                                      Discharge Superheat (F):   44.94      1.32
  0 36 WattHours Per Count  -----------------------------------------------------
        Counts :  3512.00                            Test Period (seconds):  1836.43
      WattHours:  1264.32                         Cooling EER:  13.03     COP:   4.41

 Condenser DP (psia    11 50   0 46   Evap Inlet Pressure after TXV (psia):  147.89    0.98
                                           Evap Inlet Temp after TXV (F):    52.11    0.62
                                                   Evap Inlet Temp2 (F):    51.98    0.45
                                                   Evap Inlet Temp3 (F):    52.53    0.76
                                                     Cond Inlet Temp (F):   158.76    1.11
                                                      Cond Exit Temp (F):   101.99    1.17
```

```
COOLING TEST SUMMARY  SET

        DATA FILENAME: b010328a.dat   SUMMARY FILENAME: b010328a.sum
                                                                  Range
  Air-Side Conditions        Range  Total Air-Side Capacity: 38144.46   868.97
    Indoor Dry-Bulb : 79.966    0.62    Sensible Cap (Btu/h): 28009.22   873.38
Indoor Inlet Dew (F)   60 303    0 15     Latent Cap (Btu/h): 10135.24   486.89
Indoor Exit Dry-Bulb: 59 548    0 22    EvapAir Delta T (F):  21.18     0.43
 Indoor Exit Dew (F): 56 277    0 21
  Outdoor Dry-Bulb (F)    95 016  0 48   Sensible Heat Ratio:  0 730    0 0105
  Indoor Airflow (CFM): 1186.88  18.28
 Indoor Airflow (SCFM): 1201 33  17 87      (0 075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011398
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.009629
     Barometric Pressure (in HG):  29.95      Nozzle Temp (F):  60 12   0 83
        7 inch Nozzle Pressure Drop (in Water):  1.284    0.0 3
  Evaporator Coil Air Pressure Drop (in Water):  0.150    0.033

  Refrigerant Side Conditions
   Discharge Pressure (psia):  400.48    Z.446         Ref-side Cap (Btu/h)   38301.44   5Z0.78
           Suction Pressure :  160.79    0.504          Ref-side Cap (tons):      3.19     0.04
    Condenser Inlet Pressure:  399.81    Z.594       Liq Line Mdot (lbm/min):     9.03     0.11
     Condenser Exit Pressure:  388.75    Z.443       Disch Mass Flow (lb/min)    -0.10     0.00
   Liq MassMeter Inlet (psia):  388.75    Z.443     Liq Line Density (lb/ft3):    76.07     0.42
 LiqMeter Exit/TXV In (psia):  373.70    Z.203     TXV Inlet Temperature (F):     98.40     0.56
                                                    TXV Inlet Subcooling (F):     10.37     0.67
 Evaporator Press Drop (psid):    4.70    0.367   Disch Line Density (lb/ft3):    21.510    0.408
   Evap Exit Pressure (psia):  162.39    0.487             Suction Temp (F):     57.64     1.43
   Evap Exit Temperature (F):   55.92    3.910         Suction Superheat (F):     6.17     1.45
     Evap Exit Superheat (F):    3.83    3.864           Discharge Temp (F):    159.24     0.94
                                                      Discharge Superheat (F):    45.03     0.92
  0 36 WattHours Per Count  -----
        Counts :  5320.00                              Test Period (seconds):  Z647.52
     WattHours:  1315.20                           Cooling EER:  14.63    COP    4.29

  Condenser DP (psid)    11 Z0    0 40   Evap Inlet Pressure after TXV (psia):  149.20     0.68
                                               Evap Inlet Temp after TXV (F):   52.60     0.54
                                                        Evap Inlet Temp2 (F):   52.39     0.47
                                                        Evap Inlet Temp3 (F):   53.01     0.41
                                                         Cond Inlet Temp (F):  159.16     0.65
                                                          Cond Exit Temp (F):  100.68     0.57
```

COOLING TEST SUMMARY SHEET

```
        DATA FILENAME: b010329x.dat   SUMMARY FILENAME: b010329x.sum
                                                                          Range
  Air-Side Conditions        Range  Total Air-Side Capacity: 29985.34   919.9
    Indoor Dry-Bulb : 80.111   0.80    Sensible Cap (Btu/h): 25280.38   781.1
Indoor Inlet Dew (F): 60.507   0.20      Latent Cap (Btu/h):  4704.96   548.7
Indoor Exit Dry-Bulb: 61.768   0.32     EvapAir Delta T (F):  19.14     0.51
 Indoor Exit Dew (F): 58.414   0.40
  Outdoor Dry-Bulb (F): 125.460  1.58   Sensible Heat Ratio:   0.843    0 0174
  Indoor Airflow (CFM): 1189.81  12.10
 Indoor Airflow (SCFM): 1198.98  11.80      (0.075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011236
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.010413
     Barometric Pressure (in HG):  29.95       Nozzle Temp (F):  ≤Z 20   0 57
        7 inch Nozzle Pressure Drop (in Water):  1.284    0.026
  Evaporator Coil Air Pressure Drop (in Water):  0.147    0.006
 -------------------------------------------------------------------------------
  Refrigerant Side Conditions
   Discharge Pressure (psia):  567.66    5.039        Ref-side Cap (Btu/h) : 30216.06   547.83
           Suction Pressure :  171.04    1.007          Ref-side Cap (tons):     2.52     0.05
    Condenser Inlet Pressure:  567.13    5.089      Liq Line Mdot (lbm/min):     8.71     0.13
     Condenser Exit Pressure:  557.96    5.032     Disch Mass Flow (lb/min):    -0.10     0.00
  Liq MassMeter Inlet (psia):  557.96    5.032    Liq Line Density (lb/ft3):    89.06     0.35
 LiqMeter Exit/TXV In (psia):  541.65    4.≤48    TXV Inlet Temperature (F):   126.60     1.00
                                                   TXV Inlet Subcooling (F):    11.27     0.44
 Evaporator Pres Drop (psid):    5.64    0.413  Disch Line Density (lb/ft3):    5.257    0.203
   Evap Exit Pressure (psia):  172.32    1.071             Suction Temp (F):    62.61     1.33
   Evap Exit Temperature (F):   59.75    2.733         Suction Superheat (F):     7.31     1.32
     Evap Exit Superheat (F):    3.98    2.746            Discharge Temp (F):   213.01     2.35
                                                       Discharge Superheat (F):    71.15     1.95
  0 36 WattHours Per Count  ----------------------------------------------------------------
        Counts :  5864.00                           Test Period (seconds):  1909.54
     WattHours:  2111.04                          Cooling EER:    7.53    COP:   2.21

  Condenser ΔP (psid):   9 31   0 20   Evap Inlet Pressure after TXV (psia):  160.12    1.47
                                            Evap Inlet Temp after TXV (F):   56.64    0.64
                                                     Evap Inlet Temp2 (F):   56.14    0.31
                                                     Evap Inlet Temp3 (F):   56.72    0.53
                                                      Cond Inlet Temp (F):  212.99    2.35
                                                       Cond Exit Temp (F):  131.13    1.08
```

```
COOLING TEST SUMMARY SHEET

       DATA FILENAME: b010329b.dat   SUMMARY FILENAME: b010329b.sum
                                                                       Range
 Air-Side Conditions          Range  Total Air-Side Capacity: 29930.40  1199.84
   Indoor Dry-Bulb : 80.084    0.78    Sensible Cap (Btu/h): 25446.25   891.42
Indoor Inlet Dew (D     60 222    0 05      Latent Cap (Btu/h):  4484.14   665.62
Indoor Exit Dry-Bulb  61 612    0 28     EvapAir Delta T (F):  19.25     0.63
 Indoor Exit Dew (F)  58 212    0 31
  Outdoor Dry-Bulb (D ) 125.409  0.98   Sensible Heat Ratio    0 850    0 0208
  Indoor Airflow (CFM) 1190.45  16.59
 Indoor Airflow (SCFM)  1199 87  1 54      (0 075 lb/ft3 standard air)
   Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011120
    Evap Exit Humidity Ratio (lbH2O/lbAir):   0.010336
    Barometric Pressure (in HG):  29.95      Nozzle Temp (F):  6Z 1Z   0 6Z
       7 inch Nozzle Pressure Drop (in Water):  1.286     0.036
  Evaporator Coil Air Pressure Drop (in Water):  0.147     0.009
-------------------------------------------------------------------------------

  Refrigerant Side Conditions
   Discharge Pressure (psia):  566.16     4.208          Ref-side Cap (Btu/h) : 301E7 70   610.41
           Suction Pressure :  170.61     0.629            Ref-side Cap (tons):     2.51     0.05
    Condenser Inlet Pressur :  565.75     4.404        Liq Line Mdot (lbm/min):     8.69     0.13
     Condenser Exit Pressure:  556.59     4.396        Disch Mass Flow (lb/min):   -0.10     0.00
  Liq MassMeter Inlet (psia):  556.59     4.396       Liq Line Density (lb/ft3):   89.04     0.34
 LiqMeter Exit/TXV In (psia):  540.36     4.897       TXV Inlet Temperature (F):  126.45     0.92
                                                       TXV Inlet Subcooling (F):   11.22     0.74
 Evaporator Pres Drop (psid):    5.62     0.438    Disch Line Density (lb/ft3):    5.234    0.172
   Evap Exit Pressure (psia):  171.91     0.706               Suction Temp (F):   62.40     2.66
   Evap Exit Temperature (F):   59.35     3.874          Suction Superheat (F):    7.25     2.61
     Evap Exit Superheat (F):    3.73     3.963             Discharge Temp (F):  213.19     1.88
                                                        Discharge Superheat (F)    71.55     1.65
  0 36 WattHours Per Count   ----------------------------------------------------------------------
        Counts : 10243.00                           Test Period (seconds):  3295.59
      WattHours:  3687.48                        Cooling EER:   7.43    COP:   2.18

  C  Denser DP (psid)      9 29    0 27    Evap Inlet Pressure after TXV (psia):  159.66      0.98
                                              Evap Inlet Temp after TXV (F):   56.49      0.59
                                                        Evap Inlet Temp2 (F):   56.13      0.57
                                                        Evap Inlet Temp3 (F):   56.48      0.54
                                                         Cond Inlet Temp (F):  213.15      1.82
                                                          Cond Exit Temp (F):  131.02      0.93
```

COOLING TEST SUMMARY SHEET

DATA FILENAME: b010330k.dat   SUMMARY FILENAME: b010330k.sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb	79 8≤≤	0 90	Total Air-Side Capacity:	40345.16	1161.20
Indoor Inlet Dew (F)	60 130	0 15	Sensible Cap (Btu/h):	29409.59	951.19
Indoor Exit Dry-Bulb	58 385	0 33	Latent Cap (Btu/h):	10935.57	616.92
Indoor Exit Dew (F	54 970	0 31	EvapAir Delta T (F):	22.25	0.61
Outdoor Dry-Bulb (F):	82.122	0.40	Sensible Heat Ratio	0 72З	0 015≤
Indoor Airflow (CFM):	1183.95	11.10			
Indoor Airfl (SCFM	1201 2т	11 97	(0.075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir): 0.011085
Evap Exit Humidity Ratio (lbH2O/lbAir): 0.009176
Barometric Pressure (in HG) Z9.95   Nozzle Temp (F): 59 02   0 79
7 inch Nozzle Pressure Drop (in Water): 1.280   0.025
Evaporator Coil Air Pressure Drop (in Water): 0.150   0.007

Refrigerant Side Conditions

Discharge Pressure (psia):	З42.35	1.174	Ref-side Cap (Btu/h) :	40Z31.21	80ˀ.46
Suction Pressure :	ˉ56.49	0.655	Ref-side Cap (tons):	3.36	0.07
Condenser Inlet Pressure	З41.55	1.077	Liq Line Mdot (lbm/min):	8.95	0.15
Condenser Exit Pressure	З29.10	1.221	Disch Mass Flow (lb/min):	-0.10	0.00
Liq MassMeter Inlet (psia):	З29.10	1.221	Liq Line Density (lb/ft3):	71.13	0.25
LiqMeter Exit/TXV In (psia):	З14.55	1.714	TXV Inlet Temperature (F):	88.02	0.53
			TXV Inlet Subcooling (F):	8.05	0.65
Evaporator Pres Drop (psid):	4.08	0.338	Disch Line Density (lb/ft3):	11.076	2.057
Evap Exit Pressure (psia):	158.13	0.973	Suction Temp (F)	55.86	2.95
Evap Exit Temperature (F):	54.95	5.669	Suction Superheat (F)	6.04	2.88
Evap Exit Superheat (F):	4.50	5.730	Discharge Temp (F):	141.54	1.37
			Discharge Superheat (F):	39.07	1.42

0 36 WattHours Per Count
Counts : 4618.00
WattHours: 1662.48

Test Period (seconds): 2718.65
Cooling EER: 18.33   COP: 5.37

Condenser DP (psid):	1Z т0	0 ≤5	Evap Inlet Pressure after TXV (psia):	144.43	1.17
			Evap Inlet Temp after TXV (F):	50.75	0.39
			Evap Inlet Temp2 (F):	50.57	0.35
			Evap Inlet Temp3 (F):	51.28	0.52
			Cond Inlet Temp (F):	141.35	1.47
			Cond Exit Temp (F):	89.53	0.69

COOLING TEST SUMMARY SHEET

```
        DATA FILENAME  b01033?x.dat    SUMMARY FILENAME: b010331x.sum
                                                                Range
  Air-Side Conditions          Range  Total Air-Side Capacity: 33304.65   722.43
    Indoor Dry-Bulb : 80.051    0 85    Sensible Cap (Btu/h): 26674.30   607.93
Indoor Inlet Dew (D)  60.293    0 34      Latent Cap (Btu/h):  6630.35   527.84
Indoor Exit Dry-Bulb  60.636    0 52     EvapAir Delta T (F):  20.22     0.28
 Indoor Exit Dew (F): 57.276    0 43
  Outdoor Dry-Bulb (F): 115.386  1 50   Sensible Heat Ratio    0 801    0 0141
   Indoor Airflow (CFM): 1186 35  ?3 38
 Indoor Airflow (SCFM): 1198 18  13 45      (0.075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011149
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.009989
     Barometric Pressure (in H5     29 95       Nozzle Temp (F):  61.17    0 75
        7 inch Nozzle Pressure Drop (in Water):  1.280    0.0z9
  Evaporator Coil Air Pressure Drop (in Water):  0.149    0.0o5
 ---------------------------  -------
  Refrigerant Side Conditions
   Discharge Pressure (psia):  507.82    8.024         Ref-side Cap (Btu/h) : 33303.74   481.22
           Suction Pressure :  166.96    1.662          Ref-side Cap (tons):     2.78     0.04
    Condenser Inlet Pressure:  507.14    7.879       Liq Line Mdot (lbm/min):    8.99     0.14
     Condenser Exit Pressure:  496.95    7.669      Disch Mass Flow (lb/min     -0.10     0.00
  Liq MassMeter Inlet (psia):  496.95    7.669     Liq Line Density (lb/ft3)    83.39     0.82
 LiqMeter Exit/TXV In (psia):  480.19    7.149      TXV Inlet Temperature (F   113.45     1.42
                                                     TXV Inlet Subcooling (F     9.71     0.47
 Evaporator Pres Drop (psid):    5.70    0.397   Disch Line Density (lb/ft3):    7.576   0.231
   Evap Exit Pressure (psia):  168.55    1.582             Suction Temp (F):   60.51     0.64
   Evap Exit Temperature (F):   57.88    1.545         Suction Superheat (F):    6.71     0.96
     Evap Exit Superheat (F):    3.49    1.363           Discharge Temp (F):  191.07     1.99
                                                    Discharge Sup rheat (F    58.27     0.88
  0 36 WattHours Per Count  -------------       --------------------  -------------------
        Counts :  4685.00                          Test Period (seconds)  1831.60
      WattHours:  1686.60                       Cooling EER: 10.05     COP    2.94

  Condenser DP (psia    10 34    0 30     Ev4 Inlet Pressure after TXV (psia):  156.32     2.34
                                              Evap Inlet Temp after TXV (F):   55.26     0.64
                                                       Evap Inlet Temp2 (F):   54.89     1.09
                                                       Evap Inlet Temp3 (F):   55.25     0.64
                                                        Cond Inlet Temp (F):  190.90     1.80
                                                         Cond Exit Temp (F):  122.30     1.52
```

COOLING TEST SUMMARY SHEET

```
        DATA FILENAME: b010402a.dat   SUMMARY FILENAME: b010402a.sum
                                                                    Range
  Air-Side Conditions        Range  Total Air-Side Capacity: 29006.80  1097.52
    Indoor Dry-Bulb : 79.721    0.89    Sensible Cap (Btu/h): 24642.10   443.95
Indoor Inlet Dew (F): 60.323    1.28      Latent Cap (Btu/h):  4364.70   975.40
Indoor Exit Dry-Bulb: 61.806    0.94     EvapAir Delta T (F):  18.69     0.21
 Indoor Exit Dew (F : 58.373    1.04
  Outdoor Dry-Bulb (F): 129.817  1.74   Sensible Heat Ratio    0 850    0 0280
  Indoor Airflow (CFM): 1188.17  22.42
 Indoor Airflow (SCFM): 1197.11  20.25      ( 0.075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011163
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.010398
     Barometric Pressure (in HG):  29 05      Nozzle Temp (F):  ≤2 30   1 27
        7 inch  Nozzle Pressure Drop (in Water):  1.280     0.046
  Evaporator Coil Air Pressure Drop (in Water):  0.14≤     0.009
 ---------------------------  -------
  Refrigerant Side Conditions
   Discharge Pressure (psia):  592.22    5.920        Ref-side Cap (Btu/h) : 28870.76  1054.86
           Suction Pressure :  171.66    2.367         Ref-side Cap (tons):     2.41     0.09
    Condenser Inlet Pressure:  591.66    5.921      Liq Line Mdot (lbm/min):    8.58     0.27
     Condenser Exit Pressure:  582.79    6.106     Disch Mass Flow (lb/min):   -0.10     0.10
  Liq MassMeter Inlet (psia):  582.79    6.106    Liq Line Density (lb/ft3):   90.54     0.26
 LiqMeter Exit/TXV In (psia):  566.38    7.051     TXV Inlet Temperature (F):  130.01     0.64
                                                    TXV Inlet Subcooling (F):   11.52     0.84
 Evaporator Pres Drop (psid):    5.73    0.528  Disch Line Density (lb/ft3):   7.023    0.271
   Evap Exit Pressure (psia):  1¬2.90    2.628             Suction Temp (F):   63.20     1.25
   Evap Exit Temperature (F):  ≤0.09    2.739         Suction Superheat (F):    7.67     1.21
     Evap Exit Superheat (F):    4.11    2.483            Discharge Temp (F)   221.46     3.18
                                                     Discharge Superheat (F)    76.11     2.43
  0 3≤ Wa  Hours Per Count  ------------------------------------------------
        C  nts :  6474.00                          Test Period (seconds):  1985.78
     Wat  owrs:  2330.64                        Cooling EER:   6.87    COP:   2.01

  Condenser DP (psid):    9.03   0 54   Evap Inlet Pressure after TXV (psia):  160.73     2.93
                                            Evap Inlet Temp after TXV (F):   57.01     1.03
                                                    Evap Inlet Temp2 (F):   56.51     1.20
                                                    Evap Inlet Temp3 (F):   56.93     0.94
                                                     Cond Inlet Temp (F):  221.42     3.46
                                                      Cond Exit Temp (F):  134.94     1.11
```

COOLING TEST SUMMARY SHEET

```
        DATA FILENAME: b010403a dat   SUMMARY FILENAME: b010403a.s
                                                                      Range
  Air-Side Conditions          Range  Total Air-Side Capacity: 3058[illegible].00   47[illegible] 09
    Indoor Dry-Bulb     79 701    0 28    Sensible Cap (Btu/h   25370 10    406 16
 Indoor Inlet Dew (F):  60.259    0.15      Latent Cap (Btu/h):  5215 39     300 4[illegible]
 Indoor Exit Dry-Bulb:  61.294    0.22     Evap Air Delta T (F)    19 19     0 21
  Indoor Exit Dew (F):  57.913    0.16
  Outdoor Dry-Bulb (F)   124.991   1 24   Sensible Heat Ratio:    0 829     0 0096
   Indoor Airflow (CFM)   1190.29   14 8[illegible]
  Indoor Airflow (SCFM)   1200 34   14 98     (0 075 lb/ft3 standard air)
     Evap Inlet Humidity Ratio (lbH2O/lbAir)     0 011135
      Evap Exit Humidity Ratio (lbH2O/lbAir):    0 010324
      Barometric Pressure (in HG):  29.95       Nozzle Temp (F :   [illegible]1 33    0 [illegible]6
         7 inch Nozzle Pressure Drop (in Water     1 28[illegible]     0 032
  Evaporator Coil Air Pressure Drop (in Water)    0 143     0.011
 -------------------------------------------------------------------------------

  Refrigerant Side Conditions
   Discharge Pressure (psia):  559.45     2 838        Ref-side Cap (Btu/h) : 30[illegible]24.20   47[illegible].78
            Suction Pressure :  169.38     0 428          Ref-side Cap (tons):      2.55     0.04
    C Condenser Inlet Pressure:  553 06     2 [illegible]92        Liq Line Mdot (lbm/min)     8.75     0.10
     Condenser Exit Pressure:  [illegible]49 [illegible]3     3 078     Disch Mass Flow (lb/min):    -0.10     0.00
  Liq MassMeter Inlet (psia):  549.63     3.078     Liq Line Density (lb/ft3):    88.58     0.52
 LiqMeter Exit/TXV In (psia):  532.52     3.085      TXV Inlet Temperature (F):   125.55     0.68
                                                     TXV Inlet Subcooling (F):    10.93     0.52
  Evaporator Pres Drop (psid):    5.72     0 523   Disch Line Density (lb/ft3):     7 102    0.161
   Evap Exit Pressure (psia):  170.70     0 608            Suction Temp (F)       62.06     0.79
   Evap Exit Temperature (F):   59.05     1.715        Suction Superheat (F)        7.36     0.81
     Evap Exit Superheat (F):    3.87     1 8 8            Discharge Temp (F)     208.77     0.94
                                                    Discharge Superheat (F)      [illegible]3.11     0.70
  0 36 WattHours Per Count   ----------------------------------------------------------
        Counts :   5[illegible]01 00                            Test Period (seconds):  1915.92
      WattHours    2016 3[illegible]                         Cooling EER:   [illegible]       COP:    2.37

  Condenser DP (psid)      9 49    0 18     Evap Inlet Pressure after TXV (psia):  158.61     0.49
                                              Evap Inlet Temp after TXV (F):   56.14     0.29
                                                       Evap Inlet Temp2 (F):   55.64     0.46
                                                       Evap Inlet Temp3 (F):   56.23     0.46
                                                        Cond Inlet Temp (F):  208.31     1.14
                                                         Cond Exit Temp (F):  130.15     0.47
```

COOLING TEST SUMMARY SHEET

DATA FILENAME: b010410a.dat   SUMMARY FILENAME: b010410a.sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb	80 1[illegible]5	0 1[illegible]	Total Air-Side Capacity:	33800.92	394.19
Indoor Inlet Dew (F	[illegible] [illegible]06	0 30	Sensible Cap (Btu/h):	27386.28	422.26
Indoor Exit Dry-Bulb	[illegible]0 2[illegible]	0 21	Latent Cap (Btu/h):	6414.64	280.87
Indoor Exit Dew (F	[illegible]6 [illegible]37	0 6	EvapAir Delta T (F):	20.67	0.20
Outdoor Dry-Bulb (F):	114.850	1.07	Sensible Heat Ratio	0 810	0 0076
Indoor Airfl (CFM)	11[illegible]1 05	15 76			
Indoor Airflow (SCFM)	1203 83	15 [illegible]4	(0 075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir): 0.010874
Evap Exit Humidity Ratio (lbH2O/lbAir): 0.009757
Barometric Pressure (in HG : 2[illegible] [illegible]5   Nozzle Temp (F): 60 8[illegible] 0 42
7 inch Nozzle Pressure Drop (in Water : 1.291 0.034
Evaporator Coil Air Pressure Drop (in Water : 0.148 0.006

Refrigerant Side Conditions					
Discharge Pressure (psia):	502.47	.468	Ref-side Cap (Btu/h) :	33558.49	47[illegible].88
Suction Pressure :	165.43	.655	Ref-side Cap (tons):	2.80	0.04
Condenser Inlet Pressure:	501.88	.272	Liq Line Mdot (lbm/min):	8.94	0.12
Condenser Exit Pressure:	491.82	.514	Disch Mass Flow (lb/min):	-0.10	0.00
Liq MassMeter Inlet (psia):	491.82	.514	Liq Line Density (lb/ft3):	84.78	0.30
LiqMeter Exit/TXV In (psia):	474.50	.469	TXV Inlet Temperature (F)	116.76	0.21
			TXV Inlet Subcooling (F :	10.46	0.28
Evaporator Pres Drop (psid):	5.59	0.400	Disch Line Density (lb/ft3):	6.804	0.161
Evap Exit Pressure (psia):	166.97	0.657	Suction Temp (F):	59.99	0.49
Evap Exit Temperature (F):	57.13	1.711	Suction Superheat (F):	6.75	0.52
Evap Exit Superheat (F):	3.32	1.702	Discharge Temp (F):	189.29	0.67
			Discharge Superheat (F):	57.34	0.65

0.36 WattHours Per Count
Counts : 5336.00
WattHours: 1920.96

Test Period (seconds): 2122.[illegible]2
Cooling EER: 1[illegible].37 COP 3.04

Condenser dP (psid)	10 20	0 27	Evap Inlet Pressure after TXV (psia):	154.68	0.78
			Evap Inlet Temp after TXV (F):	54.77	0.20
			Evap Inlet Temp2 (F):	54.50	0.20
			Evap Inlet Temp3 (F):	54.70	0.31
			Cond Inlet Temp (F):	189.13	0.86
			Cond Exit Temp (F):	120.55	0.41

COOLING TEST SUMMARY SHEET

```
        DATA FILENAME: b010425x.dac    SUMMARY FILENAME: b010425x.sum
                                                                         Range
  Air-Side Conditions        Range  Total Air-Side Capacity: 29413.68   486.91
     Indoor Dry-Bulb : 79 885   0.46    Sensible Cap (Btu/h): 24882.60   392.79
Indoor Inlet DP (F)    50 430   0 15      Latent Cap (Btu/h):  4531.09   346.74
Indoor Exit Dry-Bulb: 61 877   0 22     EvapAir Delta T (F):  18.80     0.15
 Indoor Exit DP (F)   58.480   0 15
  Outdoor Dry-Bulb (F): 129 905  1 19   Sensible Heat Ratio    0 845    0 0103
  Indoor Airflow (CFM): 1192 65  12 84
 Indoor Airflow (SCFM): 1201 51  13 44      (0.075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011229
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.010438
     Barometric Pressure (in HG):  29.95      Nozzle Temp (F):  62.35   0.72
        7 inch Nozzle Pressure Drop (in Water):  1.290    0.028
  Evaporator Coil Air Pressure Drop (in Water):  0.147    0.007
                                             -------------------------------------------

  Refrigerant Side Conditions
   Discharge Pressure (psia):  587.90    1.957        Ref-side Cap (Btu/h) : 232.56.35   396.70
           Suction Pressure :  171.68    0.378          Ref-side Cap (tons):      2.44     0.03
    Condenser Inlet Pressure:  587.44    1.957       Liq Line Mdot (lbm/min):     8.69     0.12
     Condenser Exit Pressure:  577.98    2.003      Disch Mass Flow (lb/min):    -0.10     0.00
  Liq MassMeter Inlet (psia):  577.98    2.003     Liq Line Density (lb/ft3):    90.42     0.27
 LiqMeter Exit/TXV In (psia):  560.32    2.203      TXV Inlet Temperature (F):  129.95     0.10
                                                   TXV Inlet Subcooling (F):    10.70     0.39
 Evaporator Pres Drop (psid):    5.87    0.438   Disch Line Density (lb/ft3):    6.819    0.139
   Evap Exit Pressure (psia):  173.02    0.487             Suction Temp (F):    63.81     0.57
   Evap Exit Temperature (F):   60.26    1.101        Suction Superheat (F):     8.27     0.68
     Evap Exit Superheat (F):    4.23    1.171           Discharge Temp (F)   218.61     0.89
                                                    Discharge Superheat (F)    73.86     0.88
  0 36 WattHours Per Count                                                                -----
        Counts : 12278.00                               Test Period (seconds):  4021.90
      WattHours:  4420.08                             Cooling EER:   7.43    COP   2.10

  Condenser DP (psid):   9 30   0 21   Evap Inlet Pressure after TXV (psia):  160.95     0.59
                                              Evap Inlet Temp after TXV (F):   57.32     0.55
                                                      Evap Inlet Temp2 (F):   56.63     0.39
                                                      Evap Inlet Temp3 (F):   57.83     0.11
                                                       Cond Inlet Temp (F):  218.55     0.98
                                                        Cond Exit Temp (F):  134.74     0.61
```

B.2 R410A System With Custom-Fabricated Compressor

Table B.2 lists the tests performed with the custom-fabricated R410A compressor and the corresponding outdoor dry-bulb temperatures.

Table B.2 R410A tests with compressor #2

Filename	Outdoor Temperature (°F)
C010712C	129.8
C010713B	94.6
C010717A	139.4
C010718A	115.2
C010719A *	150.0
C010723A	95.1
C010723C *	152.2
C010723D *	155.4

*Compressor operated above the critical point at its discharge

```
COOLING TEST SUMMARY SHEET

       DATA FILENAME  c010712c dat    SUMMARY FILENAME  c010712c.sum
                                                                  Range
 Air-Side Conditions        Range  Total Air-Side Capacity: 30478.18  765.99
   Indoor Dry-Bulb : 79.912   0.20    Sensible Cap (Btu/h): 25584.23  671.78
Indoor Inlet Dew (F): 59.922   0.20     Latent Cap (Btu/h):  4893.94  398.64
Indoor Exit Dry-Bulb: 61.278   0.11    EvapAir Delta T (F):  19.38     0.23
 Indoor Exit Dew (F)   57 [illegible]   0 15
  Outdoor Dry-Bulb (F): 12[illegible] 84[illegible]  0 83   Sensible Heat Ratio:   0 8[illegible]    0 011[illegible]
  Indoor Airflow (CFM)  1188 44  [illegible].26
 Indoor Airfl   (SCFM   11[illegible] [illegible]7  25.45     (0 075 lb/ft3 std dry air)
   Evap Inlet Humidity Ratio (lbH2O/lbAir):  0.011000
    Evap Exit Humidity Ratio (lbH2O/lbAir):  0.010144
    Barometric Pressure (in HG):  29.95      Nozzle Temp (F)   [illegible]1 71   0 48
       7 inch Nozzle Pressure Drop (in Water):  1.283    0.0 4
 Evaporator Coil Air Pressure Drop (in Water):  0.135    0.0[illegible]8
-----------------------------------------------------------------------------
 Refrigerant Side Conditions
  Discharge Pressure (psia):  582.76   2.495      Ref-side Cap (Btu  ) : [illegible]78.45   32[illegible].41
          Suction Pressure :  169.28   0.378        Ref-side Cap (tons):       2.50    0.03
   Condenser Inlet Pressure:  582.34   2.496      Liq Line [illegible] (lbm/min):       8.97    0.09
    Condenser Exit Pressure:  571.83   2.491     Disch Mass Flow (lb/min):      -0.10    0.00
 Liq MassMeter Inlet (psia):  571.83   2.491    Liq Line Density (lb/ft3):      90.29    0.18
LiqMeter Exit/TXV In (psia):  551.69   2.840    TXV Inlet Temperature (F)     130.74    0.30
                                                 TXV Inlet Subcooling (F :       8.63    0.41
Evaporator Pres Drop (psid):    6.44   0.528   Disch Line Density (lb/ft3):      7.806   0.275
  Evap Exit Pressure (psia):  170.81   0.268            Suction Temp (F):      62.92    0.86
  Evap Exit Temperature (F):   59.88   [illegible].598        Suction Superheat (F :       8.26    0.84
    Evap Exit Superheat (F):    4.66   [illegible].652          Discharge Temp (F :     2[illegible]8.68    0.83
                                                   Discharge Superheat (F :     [illegible]4.67    0.58
 0.36 WattHours Per Count  --------------------------------------------------
       Counts :  6764.00                      Test Period (seconds):  2212.34
     WattHours:  2435.04                     Cooling EER:   7.69   COP:   2.25

 C[illegible]denser DP (psia):  10 3[illegible]   0.2[illegible]    [illegible] Inlet Pressure after TXV (psia):  177.56    0.98
                                       Evap Inlet Temp after TXV (F):   56.66    0.28
                                                Evap Inlet Temp2 (F):   55.90    0.33
                                                Evap Inlet Temp3 (F):   56.19    0.63
                                                 Cond Inlet Temp (F):  208.67    0.74
                                                  Cond Exit Temp (F):  135.57    0.59
```

COOLING TEST SUMMARY SHEET

```
      DATA FILENAME: c010713b.dat   SUMMARY FILENAME: c010713b.sum
                                                                  Range
 Air-Side Conditions        Range  Total Air-Side Capacity: 38337.24   521.04
   Indoor Dry-Bulb : 80.100   0.27    Sensible Cap (Btu/h): 28684.57   572.14
Indoor Inlet Dew (F)  [illegible]0 09[illegible]   0 [illegible]      Latent Cap (Btu/h):  9652.66   655.05
Indoor Exit Dry-Bulb: 59.174   0.22     EvapAir Delta T (F):  21.68     0.37
 Indoor Exit Dew (F   55 579   0 2[illegible]
  Outdoor Dry-Bulb (F   [illegible]4 550  0 [illegible]1   Sensible Heat Ratio    0 74[illegible]    0 01[illegible]3
  Indoor Airflow (CFM   118[illegible] 26  11 87
 Indoor Airflow (SCFM): 1202.13  11.95      (0.075 lb/ft3 standard air)
   Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011068
    Evap Exit Humidity Ratio (lbH2O/lbAir):   0.009385
    Barometric Pressure (in HG):  29.95       Nozzle Temp (F):  59.61   0 26
       7 inch Nozzle Pressure Drop (in Water):  1.284    0.026
  Evaporator Coil Air Pressure Drop (in Water):  0.139    0.008
 -------------------------
  Refrigerant Side Conditions
   Discharge Pressure (psia):  397.24    1.027          Re -side Cap (Btu/h)   [illegible]80[illegible]2.41   [illegible]02.65
           Suction Pressure :  159.73    0.629            Ref-side Cap (tons):      3.17      0.03
    Condenser Inlet Pressure:  396.61    1.223        Liq Line Mdot (lbm/min):      9.03      0.09
     Condenser Exit Pressure:  384.38    1.075       Disch Mass Flow (lb/min):     -0.10      0.00
  Liq MassMeter Inlet (psia):  384.38    1.075      Liq Line Density (lb/ft3):     76.57      0.14
 LiqMeter Exit/TXV In (psia):  366.83    1.469      TXV Inlet Temperature (F):     99.59      0.45
                                                     TXV Inlet Subcooling (F):      7.79      0.59
 Evaporator Pres Drop (psid):    4.71    0.428    Disch Line Density (lb/ft3):     8.470     0.276
   Evap Exit Pressure (psia):  161.20    0.584              Suction Temp (F):     57.49      0.61
   Evap Exit Temperature (F):   56.25    1.714         Suction Superheat (F):      6.42      0.73
     Evap Exit Superheat (F):    4.62    1.668            Discharge Temp (F :    156.47      0.84
                                                  Di scharge Superheat (F)      42.87      0.80
  0 36 WattHours Per Count  ----------------                                    ---
        Counts :  3522.00                           Test Period (seconds):  1705.32
      WattHours:  1267.92                        Cooling EER:  14.32    COP:    4.20

  C  Denser DP (psid    12 34   0.36   Evap Inlet Pressure after TXV (psia):  166.59     0.49
                                              Evap Inlet Temp after TXV (F):   52.36     0.07
                                                     Evap Inlet Temp2 (F):   52.03     0.35
                                                     Evap Inlet Temp3 (F):   52.36     0.56
                                                      Cond Inlet Temp (F):  156.37     0.73
                                                       Cond Exit Temp (F):  102.28     0.23
```

```
COOLING TEST SUMMARY SHEET

        DATA FILENAME: c010717a.dat   SUMMARY FILENAME: c010717a.sum
                                                                   Range
  Air-Side Conditions        Range  Total Air-Side Capacity: 28057.91  719.83
    Indoor Dry-Bulb : 80.018   0.22    Sensible Cap (Btu/h): 24701.23  468.77
Indoor Inlet Dew (F)   50 180   0 17     Latent Cap (Btu/h):  3356.68  310.79
Indoor Exit Dry-Bulb  62 141   0 22    EvapAir Delta T (F):  18.68     0.20
 Indoor Exit Dew (F)   58 587   0 15
  Outdoor Dry-Bulb (F): 139.377  1.11   Sensible Heat Ratio:   0 880    0 0102
  Indoor Airflow (CFM): 1191.96  17.60
 Indoor Airflow (SCFM)  1200 21  18 62     (0 075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):  0.011103
     Evap Exit Humidity Ratio (lbH2O/lbAir):  0.010517
     Barometric Pressure (in HG):  29.95    Nozzle Temp (F):  62 60   0 59
        7 inch Nozzle Pressure Drop (in Water):  1.288    0.039
  Evaporator Coil Air Pressure Drop (in Water):  0.131    0.009
  ----------------------------------------------------------------------------------

  Refrigerant Side Conditions
   Dis  arge Pressure (psia):  646.17    4.990        Ref-side Cap (Btu/h   27604.98   466.19
          Sucti   Pressure :  173.80    0.504          Ref-side Cap (tons):      2.30     0.04
    Condenser Inlet Pressure:  645.82    4.894      Liq Line Mdot (lbm/min):      8.86     0.12
     Condenser Exit Pressure:  635.82    4.641     Disch Mass Flow (lb/min):     -0.12     0.24
  Liq MassMeter Inlet (psia):  635.82    4.641    Liq Line Density (lb/ft3):     93.36     0.47
 LiqMeter Exit/TXV In (psia):  615.34    4.995    TXV Inlet Temperature (F):   137.64     0.64
                                                    TXV Inlet Subcooling (F):     10.78     0.43
 Evaporator Pres Drop (psid):     6.61    0.474  Disch Line Density (lb/ft3):     7.162    0.558
   Evap Exit Pressure (psia):  175.08    0.608            Suction Temp (F):     64.59     0.99
   Evap Exit Temperature (F):   61.16    1.320        Sucti   Superheat (F)      8.29     0.95
     Evap Exit Superheat (F):    4.40    1.302         Discharge Temp (F)    226.98     2.24
                                                    Discharge Superheat (F)     74.33     1.78
  0 36 WattHours Per Count   ---------------------------------------------------------
        Counts :  6678.00                          Test Period (seconds):  1841.32
      WattHours:  2404.08                       Cooling EER :  5.97    COP    1.75

  Condenser DP (psid)    0 83   0 22    Ev  Inlet Pressure after TXV (psia):  181.95     0.98
                                           Evap Inlet Temp after TXV (F):   57.96     0.20
                                                    Evap Inlet Temp2 (F):   57.53     0.55
                                                    Evap Inlet Temp3 (F):   57.92     0.41
                                                     Cond Inlet Temp (F):  227.00     2.15
                                                      Cond Exit Temp (F):  142.96     0.57
```

COOLING TEST SUMMARY SHEET

DATA FILENAME: c010718a.dat SUMMARY FILENAME: c010718a.sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb	≤0 119	0 34	Total Air-Side Capacity:	34813.10	525.62
Indoor Inlet Dew (F	≤0 704	0 20	Sensible Cap (Btu/h):	26912.99	419.80
Indoor Exit Dry-Bulb	60 501	0 17	Latent Cap (Btu/h):	7900.11	445.84
Indoor Exit Dew (F :	57 134	0 19	EvapAir Delta T (F):	20.35	0.17
Outdoor Dry-Bulb (F	115 240	0 52	Sensible Heat Ratio	0 773	0 0110
Indoor Airflow (CFM):	1188.91	14.05			
Indoor Airflow (SCFM :	1201 19	14 ≤9	(0 075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir): 0.011316

Evap Exit Humidity Ratio (lbH2O/lbAir): 0.009937

Barometric Pressure (in HG): 29.95 Nozzle Temp (F): 6 01 0 ≤0

7 inch Nozzle Pressure Drop (in Water): 1.286 0.031

Evaporator Coil Air Pressure Drop (in Water): 0.137 0.010

--------------------------------- ------

Refrigerant Side Conditions					
Discharge Pressure (psia):	505.64	1.223	Ref-side Cap (Btu/h) :	34327.65	3≡0.99
Suction Pressure :	166.62	0.478	Ref-side Cap (tons):	2.86	0.03
Condenser Inlet Pressure:	505.08	1.223	Liq Line Mdot (lbm/min):	9.16	0.10
Condenser Exit Pressure:	494.14	1.221	Disch Mass Flow (lb/min):	-0.10	0.00
Liq MassMeter Inlet (psia):	494.14	1.221	Liq Line Density (lb/ft3):	84.78	0.31
LiqMeter Exit/TXV In (psia):	474.50	1.224	TXV Inlet Temperature (F):	117.05	0.48
			TXV Inlet Subcooling (F):	10.16	0.46
Evaporator Pres Drop (psid):	5.79	0.379	Disch Line Density (lb/ft3):	8.510	0.225
Evap Exit Pressure (psia):	168.17	0.608	Suction Temp (F):	60.05	0.62
Evap Exit Temperature (F):	57.87	1.122	Suction Superheat (F):	6.37	0.66
Evap Exit Superheat (F):	3.62	1.202	Discharge Temp (F):	185.31	0.48
			Discharge Superheat (F):	52.86	0.59

0.36 WattHours Per Count ---------

Counts : 4941.00 Test Period (seconds): 1846.38

WattHours: 1778.76 Cooling EER: 10.04 COP 2.94

Condenser DP (psid :	10 95	0 29	Evap Inlet Pressure after TXV (psia):	174.41	0.49
			Evap Inlet Temp after TXV (F):	55.23	0.22
			Evap Inlet Temp2 (F):	55.02	0.31
			Evap Inlet Temp3 (F):	55.31	0.64
			Cond Inlet Temp (F):	185.25	0.40
			Cond Exit Temp (F):	120.72	0.60

```
COOLING TEST SUMMARY SHEET

       DATA FILENAME  c010719a.dac    SUMMARY FILE NAME  c010719a.sum
                                                                  Range
  Air-Side Conditions        Range  Total Air-Side Capacity: 24801.22  511.30
    Indoor Dry-Bulb : 79.806   0.64    Sensible Cap (Btu/h): 22519.36  832.95
Indoor Inlet Dew (F): 60.939   0.60      Latent Cap (Btu/h):  2281.86  825.06
Indoor Exit Dry-Bulb: 63.561   0.37    EvapAir Delta T (F):  17.03     0.47
 Indoor Exit Dew (F): 59.956   0.44
  Outdoor Dry-Bulb (F): 150.003  0.81   Sensible Heat Ratio    0 903    0 0318
  Indoor Airfl    (CFM   1134 03  15 33
 Indoor Airflow (SCFM)  1199 20  17 20      (0.075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011413
    Evap Exit Humidity Ratio (lbH2O/lbAir):   0.011014
    Barometric Pressure (in HG):  29.95      Nozzle Temp (F):  83 82   0 77
       7 inch Nozzle Pressure Drop (in Water):  1.289    0.0 5
  Evaporator Coil Air Pressure Drop (in Water):  0.129    0.039
---------------------------------------------------------------------------------
  Refrigerant Side Conditions
   Discharge Pressure (psia):  709.29    3.180       Ref-side Cap (Btu/h) : 24159.09   407.24
           Suction Pressure :  178.49    0.982         Ref-side Cap (tons):     2.01     0.04
    Condenser Inlet Pressure:  709.12    3.328     Liq Line Mdot (lbm/min):     8.62     0.17
     Condenser Exit Pressure:  699.44    3.761    Disch Mass Flow (lb/min):     0.58     0.95
  Liq MassMeter Inlet (psia):  699.44    3.761    Liq Line Density (lb/ft3):   96.36     0.51
 LiqMeter Exit/TXV In (psia):  678.65    3.477    TXV Inlet Temperature (F):  146.27     0.63
                                                   TXV Inlet Subcooling (F):   10.46     0.57
 Evaporator Pres Drop (psid):    6.84    0.397  Disch Line Density (lb/ft3):    7.851    0.650
   Evap Exit Pressure (psia):  179.29    1.095            Suction Temp (F):   67.29     0.86
   Evap Exit Temperature (F):   63.78    1.317       Suction Superheat (F):    9.31     0.80
     Evap Exit Superheat (F):    5.52    1.180          Discharge Temp (F):   248.42     0.97
                                                    Discharge Superheat (F):   87.82     0.90
  0 36 WattHours Per Count  ---------------------------------------------------------------
       Counts :  8533.00                         Test Period (seconds):  2084.54
     WattHours:  3071.88                      Cooling EER:   4.63    COP    1.36

  Condenser DP (psid)    9 59   0 31   Evap Inlet Pressure after TXV (psia):  186.75    1.86
                                            Evap Inlet Temp after TXV (F):   59.46    0.55
                                                    Evap Inlet Temp2 (F):   58.97    0.59
                                                    Evap Inlet Temp3 (F):   59.46    0.52
                                                     Cond Inlet Temp (F):  248.60    0.91
                                                      Cond Exit Temp (F):  152.29    0.31
```

```
COOLING TEST  SUMMARY SHEET

        DATA FILENAME: c010723a.dat   SUMMARY FILENAME: c010723a.sum
                                                                      Range
  Air-Side Conditions          cope  Total Air-Side Capacity: 38722.13   604.35
    Indoor Dry-Bulb : 79.847   0.97    Sensible Cap (Btu/h): 28608.94   564.08
Indoor Inlet Dew (F)  60 283   0 47      Latent Cap (Btu/h): 10113.19   518.69
Indoor Exit Dry-Bulb  53 085   0 61     EvapAir Delta T (F):  21.56     0.39
 Indoor Exit Dew (F): 55.590   0.50
  Outdoor Dry-Bulb (F):  95.064  0.58   Sensible Heat Ratio:   0 733    0 0121
  Indoor Airflow (CFM   1183 62   1s 82
 Indoor Airflow (SCFM)  1205 53   17 88      (0.075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir):   0.011147
     Evap Exit Humidity Ratio (lbH2O/lbAir):   0.009389
     Barometric Pressure (in HG)   23.95       Nozzle Temp (F):  53 62   0 68
        7 inch Nozzle Pressure Drop (in Water):  1.291    0.037
  Evaporator Coil Air Pressure Drop (in Water):  0.135    0.008
 -------------------------------       ----------
  Refrigerant Side Conditions
   Discharge Pressure (psia):  401.76    1.272        Ref-side Cap (Btu/h)  38981.29   664.91
           Suction Pressure :  160.18    1.334         Ref-side Cap (tons):      3.19     0.06
    Condenser Inlet Pressure:  401.05    1.223      Liq Line Mdot (lbm/min):      9.04     0.15
     Condenser Exit Pressure:  389.35    1.465     Disch Mass Flow (lb/min):     -0.10     0.00
  Liq MassMeter Inlet (psia):  389.35    1.465    Liq Line Density (lb/ft3):     76.21     0.21
 LiqMeter Exit/TXV In (psia):  371.37    1.861     TXV Inlet Temperature (T :     98.71     0.56
                                                   TXV I let Subcooling (T)      9.59     0.56
 Evaporator Pres Drop (psid):    4.68    0.495  Disch Line Density (lb/ft3):     9.559    1.107
   Evap Exit Pressure (psia):  161.72    1.339            Suction Temp (F):     57.39     0.96
   Evap Exit Temperature (F):   55.73    2.067        Suction Superheat (F):      6.15     0.59
     Evap Exit Superheat (F):    3.89    1.974           Discharge Temp (F):    157.86     0.87
                                                     Discharge Superheat (F):     43.40     0.80
  0 36 WattHours Per Count
       Counts :  4089.00                           Test Period (seconds):   1835.33
     WattHours:  1472.04                         Cooling EER:  13.41    COP   3.93

  C  Denser DP (psid)   11 33   0 37    Evap Inlet Pressure after TXV (psia):  166.79    1.56
                                             Evap Inlet Temp after TXV (F):   52.37    0.50
                                                     Evap Inlet Temp2 (F):   52.41    0.82
                                                     Evap Inlet Temp3 (F):   52.82    0.50
                                                      Cond Inlet Temp (F):  157.75    0.74
                                                       Cond Exit Temp (F):  101.04    0.59
```

COOLING TEST SUMMARY SHEET

DATA FILENAME c010723c dac [redacted] FILENAME c0107Z3c sum

Air-Side Conditions		Range			Range
Indoor Dry-Bulb	80 4Z4	1.33	Total Air-Side Capacity:	24171.86	1666.44
Indoor Inlet Dew (F	≤0 582	1 03	Sensible Cap (Btu/h):	22986.59	838.14
Indoor Exit Dry-Bulb	≤3 811	1 53	Latent Cap (Btu/h):	1185.27	1485.20
Indoor Exit Dew (F	≤0 0≤8	1 23	EvapAir Delta T (F):	17.44	0.59
Outdoor Dry-Bulb (F	152 130	*T8	Sensible Heat Ratio	0 351	0 0≤04
Indoor Airflow (CFM):	1191.11	18.41			
Indoor Airflow (SCFM	1135 33	18 56	(0.075 lb/ft3 standard air)		

Evap Inlet Humidity Ratio (lbH2O/lbAir): 0.011267

Evap Exit Humidity Ratio (lbH2O/lbAir): 0.011059

Barometric Pressure (in HG): 29.95 Nozzle Temp (F): 64.17 1 34

7 inch Nozzle Pressure Drop (in Water): 1.282 0.040

Evaporator Coil Air Pressure Drop (in Water): 0.125 0.011

Refrigerant Side Conditions

Discharge Pressure (psia):	722.06	29.697	Ref-side Cap (Btu/h) :	Z3236 10	Z383.93
Suction Pressure :	179.68	4.683	Ref-side Cap (tons):	1.94	0.20
Condenser Inlet Pressure:	721.73	29.949	Liq Line Mdot (lbm/min):	8.55	0.48
Condenser Exit Pressure:	712.14	29.847	Disch Mass Flow (lb/min):	0.14	1.43
Liq MassMeter Inlet (psia):	712.14	29.847	Liq Line Density (lb/ft3):	96.63	1.90
LiqMeter Exit/TXV In (psia):	691.36	30.555	TXV Inlet Temperature (F):	148.27	4.14
			TXV Inlet Subcooling (F):	10.04	0.83
Evaporator Pres Drop (psid):	6.75	0.684	Disch Line Density (lb/ft3):	7.349	1.513
Evap Exit Pressure (psia):	180.80	4.551	Suction Temp (F):	68.17	1.38
Evap Exit Temperature (F):	64.03	2.100	Suction Superheat (F):	9.77	0.86
Evap Exit Superheat (F):	5.23	1.305	Discharge Temp (F):	252.27	9.84
			Discharge Superheat (F):	90.14	6.67

0 36 WattHours Per Count

Counts : 17372.00

WattHours: 6253.92

Test Period (seconds): 4004.5≤

Cooling EER: 4.30 COP: 1.2≤

Condenser DP (psid)	3 41	0 ≤1	Evap Inlet Pressure after TXV (psia):	188.03	4.59
			Evap Inlet Temp after TXV (F):	60.03	1.82
			Evap Inlet Temp2 (F):	59.54	1.67
			Evap Inlet Temp3 (F):	59.73	1.56
			Cond Inlet Temp (F):	252.46	9.92
			Cond Exit Temp (F):	154.07	4.12

```
COOLING TEST SUMMARY SHEET

        DATA FILENAME: c010723d.dat    SUMMARY FILENAME: c010723d.sum
                                                                      Range
  Air-Side Conditions          Range  Total Air-Side Capacity: 22698.83   880.31
    Indoor Dry-Bulb : ±0.359    0.24    Sensible Cap (Btu/h): 22644.59   674.44
Indoor Inlet Dew (F): ±0.114    0.34      Latent Cap (Btu/h):    54.24   349.79
Indoor Exit Dry-Bulb  ±3 374    0 33     EvapAir Delta T (F):  17.19     0.34
 Indoor Exit D  (F :  ±0 091    0 00
  Outdoor D -Bulb (F)  155 3    0 99   Sensible Heat Ratio:    .998     0.0152
  Indoor Airflow (CFM)  1191 22  1  15
 Indoor Airflow (SCFM): 1195 2   1  24      ( 0 075 lb/ft3 standard air)
    Evap Inlet Humidity Ratio (lbH2O/lbAir ):  0.011077
     Evap Exit Humidity Ratio (lbH2O/lbAir ):  0.011067
     Barometric Pressure (in HG):  29.95      Nozzle Temp (F):  64.28    0 5
        inch   zzle Pressure Drop (in Water):  1.282    0.0
  Evaporator Coil Air Pressure Drop (in Water):  0.122    0.036

  Refrigerant Side Conditions
   Discharge Pressure (psia):  743.08    5.773        Ref-side Cap (Btu/h) : 21530.12  2833.20
           Suction Pressure :  181.22    1.385         Ref-side Cap (tons):      1.79     0.24
    Condenser Inlet Pressure:  742.85    5.726       Liq Line Mdot (lbm/min):    8.26     1.12
     Condenser Exit Pressure:  733.54    5.715      Disch Mass Flow (lb/min):   -0.07     0.50
  Liq MassMeter Inlet (psia):  733.54    5.715     Liq Line Density (lb/ft3):   96.25    15.30
 LiqMeter Exit/TXV In (psia):  713.21    5.680     TXV Inlet Temperature (F):  151.06     1.06
                                                    TXV Inlet Subcooling (F):    9.94     0.63
 Evaporator Pres Drop (psid):    6.54    0.485   Disch Line Density (lb/ft3):   7.099    0.917
   Evap Exit Pressure (psia):  182.23    1.460             Suction Temp (F):   68.83     0.62
   Evap Exit Temperature (F):   64.43    1.205         Suction Superheat (F):    9.89     0.86
     Evap Exit Superheat (F):    5.13    1.348            Discharge Temp (F):  261.45     4.12
                                                     Discharge Superheat (F):   96.83     3.62
  0 36 WattHours Per Count
        Counts :  8917.00                           Test Period (seconds):  1838.14
     WattHours:  3210.12                          Cooling EER:   3.61    COP   1.06

  Condenser DP (psia)     3 10    0 28   Evap Inlet Pressure after TXV (psia):  189.29    1.37
                                            Evap Inlet Temp after TXV (F):   60.53    0.51
                                                    Evap Inlet Temp2 (F):   60.01    0.62
                                                    Evap Inlet Temp3 (F):   60.19    0.51
                                                     Cond Inlet Temp (F):  261.61    4.12
                                                      Cond Exit Temp (F):  156.95    1.06
```

APPENDIX C. CAPACITY AND EER UNCERTAINTY

Table C.1 gives an example of the error associated with EER and air-side capacity for several tests. A high capacity and a low capacity test were selected. Uncertainty values for capacity and EER for all tests are bounded by these values. A complete examination of error propagation for systems tested according to ASHRAE Standard 37-1988 may be found in Payne and Domanski (2001).

Table C.1 Measurement uncertainty

Filename		Value	Percent Uncertainty at a 95 % Confidence Limit on the Mean
R22 A010111a.dat	EER	18.29 ± 0.65 Btu/Wh	3.5
	Capacity	40200 ± 1171 Btu/h	2.9
R410A C010723d.dat	EER	3.61 ± 0.20 Btu/Wh	5.4
	Capacity	22699 ± 1142 Btu/h	5.0

APPENDIX D. EVAP-COND INSTRUCTION PAGES

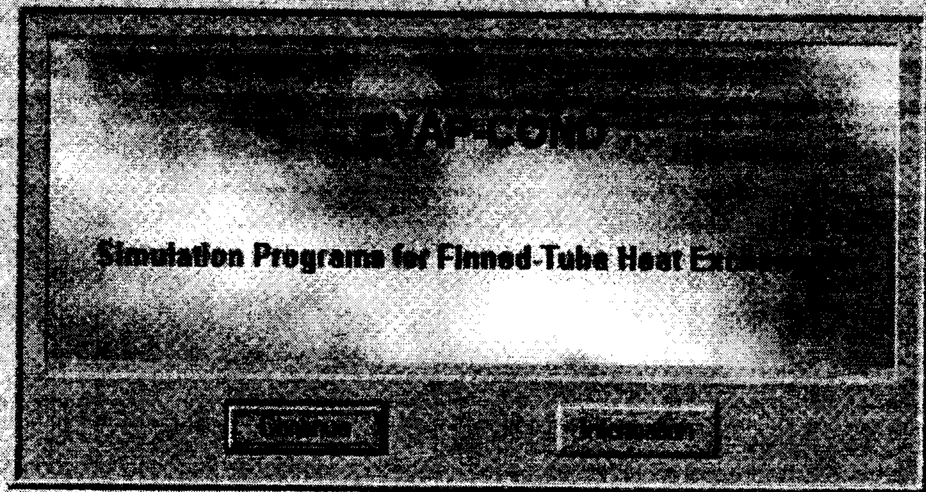
The attached pages contain **EVAP-COND** instructions prepared to facilitate the use of the EVAP-COND package.

EVAP- COND INSTRUCTIONS

EVAP-COND is a software package that contains NIST's simulation models for a finned-tube evaporator (**EVAP5**) and condenser (**COND5**). The following pages provide basic instructions on how to use this package. The instructions include preparation of input data, execution of the program, and examination of simulation results.

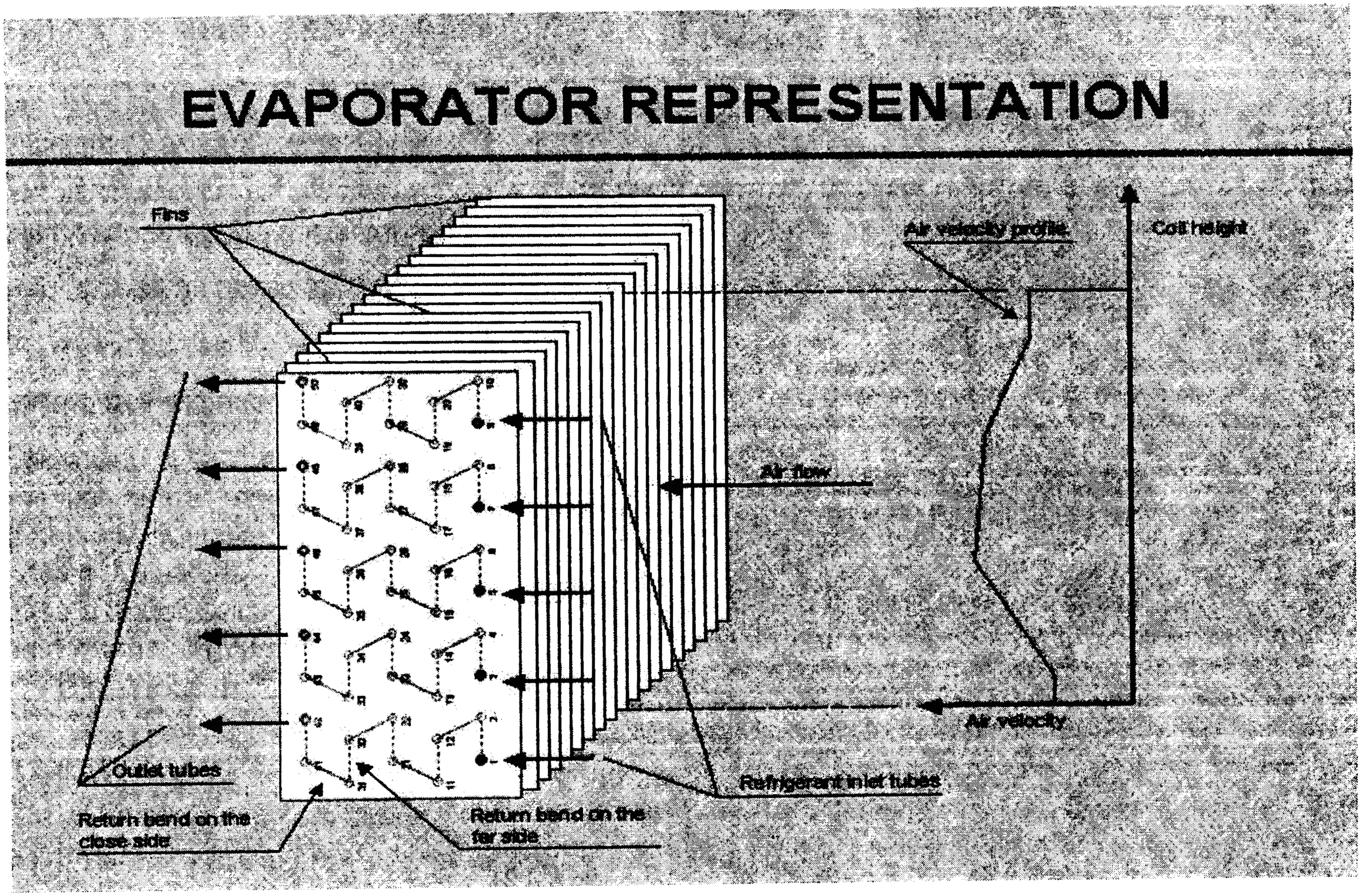
Capabilities include:

- tube-by-tube simulation
- non-uniform air distribution
- simulation of refrigerant distribution
- condenser model capable to simulate above the critical point
- 10 refrigerants and refrigerant mixtures
- REFPROP6 refrigerant properties



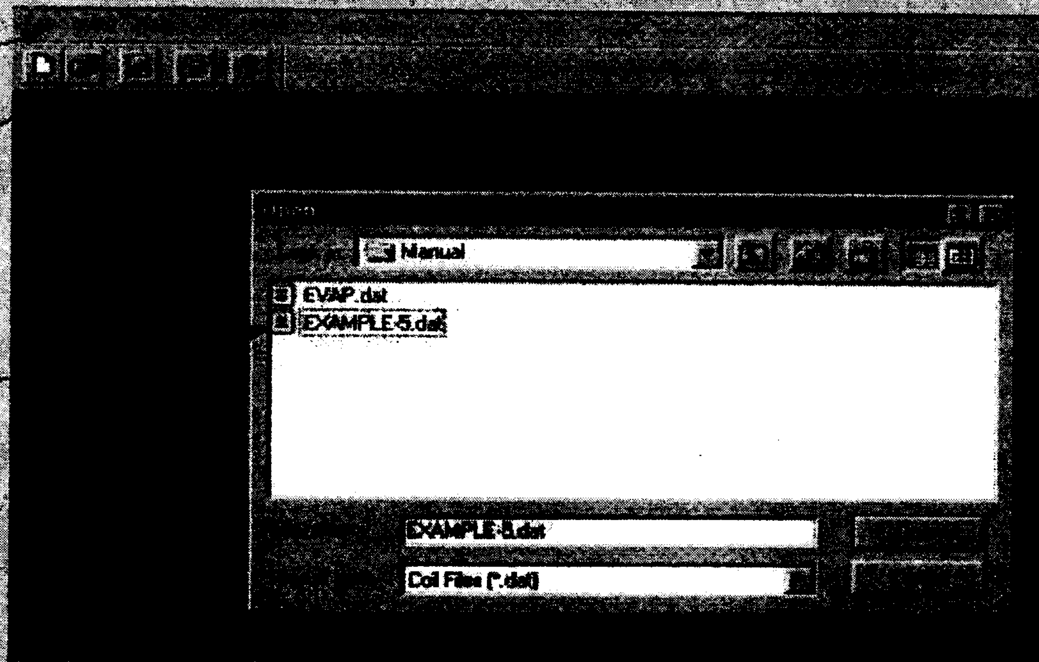
Piotr A. Domanski
National Institute of Standards and Technology
Building and Fire Research Laboratory
Gaithersburg, MD, USA

Version 1.0, October 31, 2001


EVAPORATOR REPRESENTATION
Fins
Air velocity profile
Coil height
Air flow
Air velocity
Outlet tubes
Refrigerant inlet tubes
Return bend on the close side
Return bend on the far side

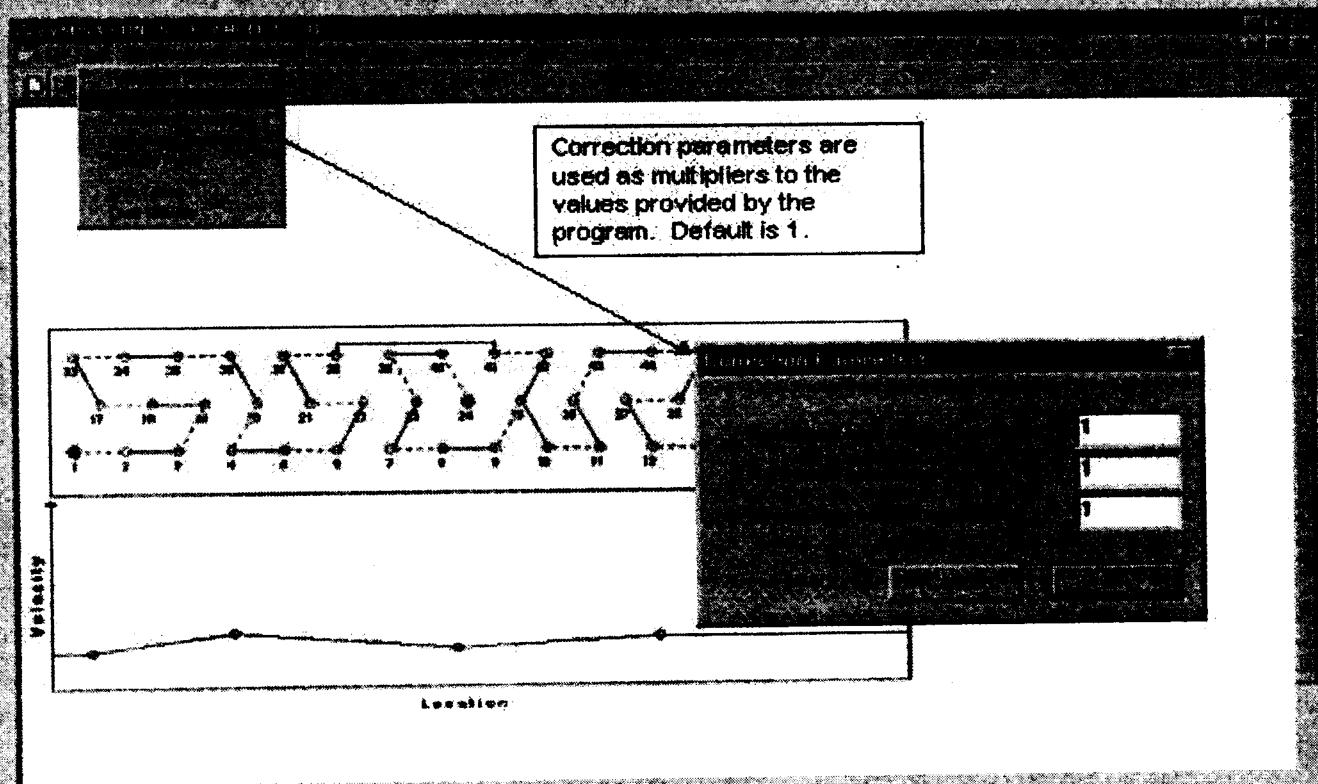
LOADING A FILE

After starting the program, click on the "open file" button on the power bar or select "Open" on the pull down menu



Open file EXAMPLE-5 to continue this tutorial.

PREPARING A SIMULATION RUN

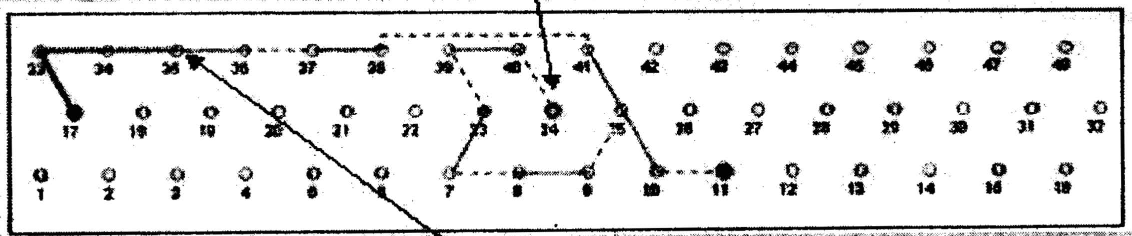


REFRIGERANT CIRCUIT DESIGN

Follow the steps below to design a circuitry:

1. place the pointer on the tube
2. press the left mouse button
3. drag the pointer to the next tube
4. release the left botton.

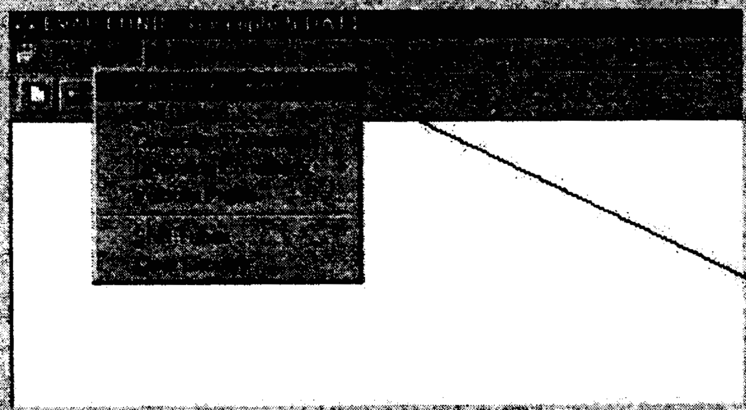
Evaporator inlet tube



Option:

You may remove a part of the circuit by positioning the pointer on a given tube and clicking twice with the left button.

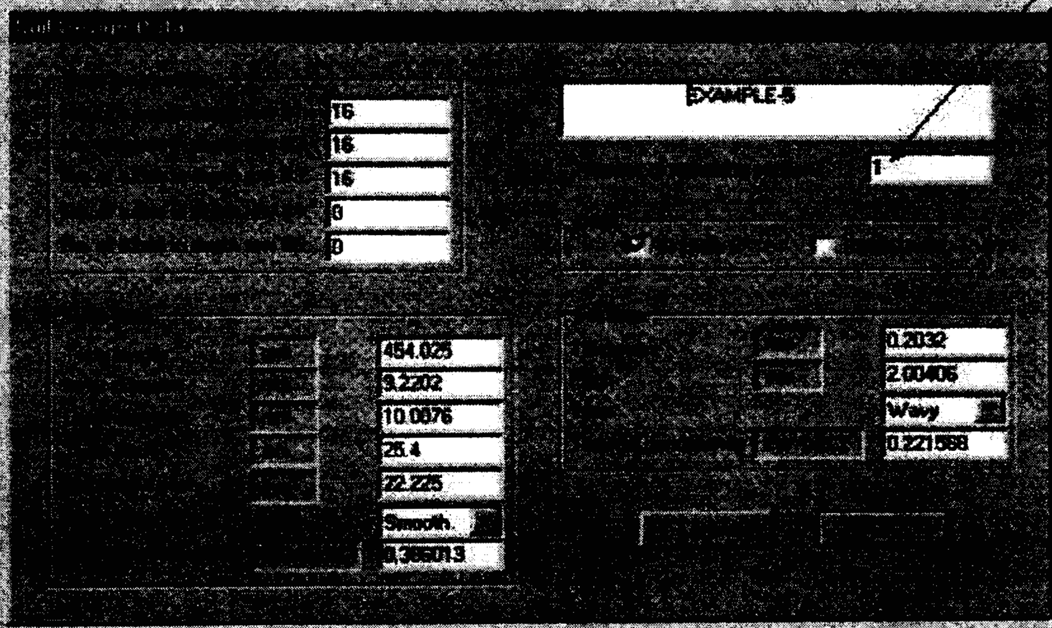
REFRIGERANT SELECTION



EXAMPLE-5 was set up to use refrigerant inlet pressure and quality as input data (see Operating Conditions). Since the specified inlet pressure is for R22, use the pull-down menu to select R22.

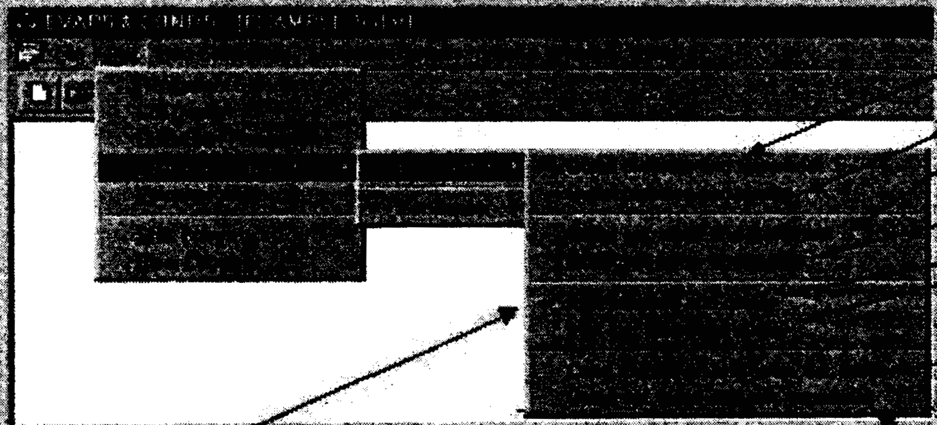
COIL DESIGN DATA

Value 1 must be used for this version of EVAP-COND.



EVAPORATOR OPERATING CONDITIONS

(Eight options)



Input includes estimated refrigerant mass flow rate, which is later iterated by EVAP5.

Inlet quality is imposed at 0.20.

Input includes refrigerant mass flow rate.

This option includes a refrigerant distributor. Condenser subcooling and evaporator superheat are imposed to be 5 °C (9 °F).

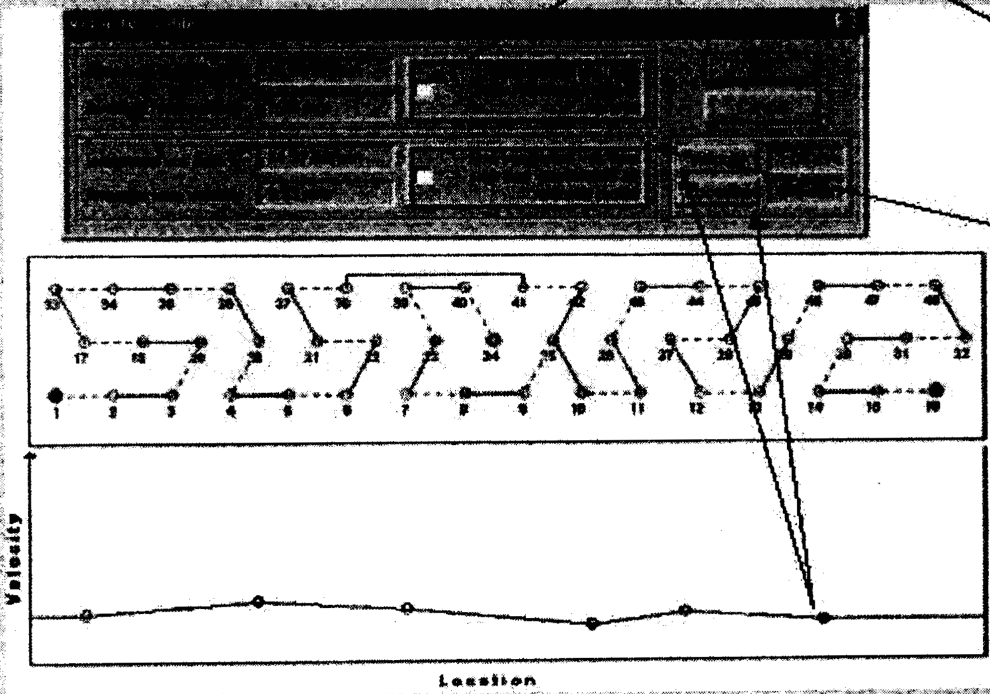
Input includes estimated refrigerant mass flow rate, which is later iterated so the least superheated tube has the specified minimum superheat.

Inlet quality is imposed at 0.20.

The mark indicates the currently selected option.

All input options include air inlet temperature, pressure, relative humidity, and volumetric flow rate.

AIR VELOCITY PROFILE



Recall that the operating conditions input data included the volumetric flow rate of air. Before you leave this window you must decide between the two options:

- to use the previously specified vol. flow rate and have the velocity profile scaled to match that value (recommended)

or

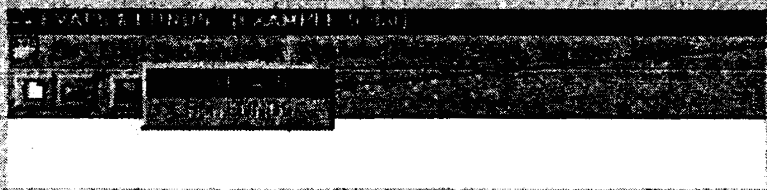
- to disregard the previously specified vol. flow rate and calculate a new volumetric flow rate from the velocity profile

To specify velocity at a given point, place the pointer at that point and click with the left button. The window above shows the coordinates. Up to sixteen velocity points may be specified.

Click on a given point to remove it.

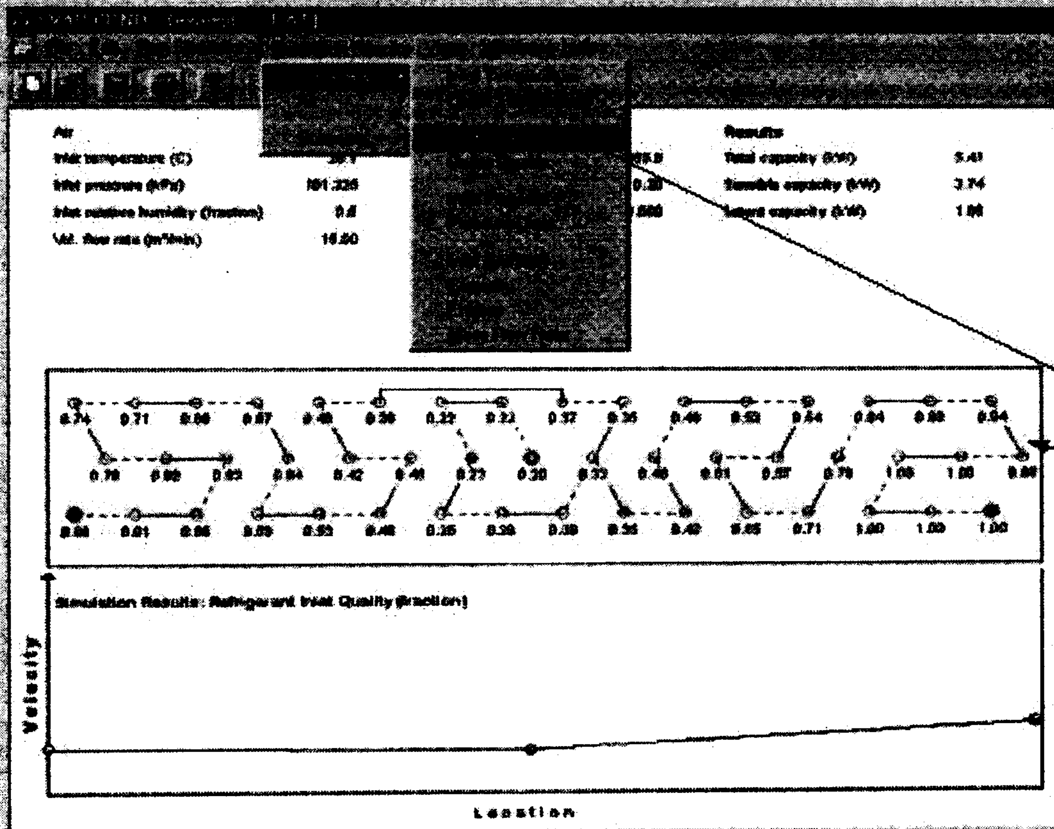
The Velocity Profile window must be open to make air velocity profile changes.

EXECUTION OF EVAP5



Use the pull-down menu to execute EVAP5.

SIMULATION RESULTS



Global simulation results are summarized in the background, under the pull-down menu Summary, and in files sizes and tt.res located in the EVAP-COND directory.

Details of refrigerant and air parameters can be displayed on the screen for individual tubes.

Refrigerant inlet quality for individual tubes

SIMULATION SUMMARY

Simulation Results Summary

```
--------------------EVAP5 SIMULATION SUMMARY------------------

Coil ID: EXAMPLE-5

REFRIGERANT: R22

REFRIGERANT SIDE
Refrigerant mass flow rate:             120.6    [kg/h]
Sensible capacity:                        3.735  [kW]
Latent capacity:                          1.677  [kW]
Total capacity:                           5.413  [kW]
Outlet saturated temp. and superheat:    7.0       6.2     [C]
Inlet and outlet temperatures:            8.7      13.3     [C]
Inlet and outlet pressures:             655.0     622.9     [kPa]
Inlet and outlet qualities:               0.200     1.000

AIR SIDE
Air volumetric flow rate:                16.0    [m^3/min]
Air temperature distribution [C]:        26.7   21.0  17.8  13.7
Air humidity    distribution [%]:       .5008  .6638 .7411 .8877

CONDITION OF REFRIGERANT LEAVING OUTLET TUBES
Tube     Quality    Temperature    Superheat    Ref. M. Fract.
  #        (-)          (C)           (C)            (-)
```

The Simulation Results Summary window displays results in the units selected for the input data. Files si.res and ui.res have the same information in the SI and Inch-Pound units, respectively.

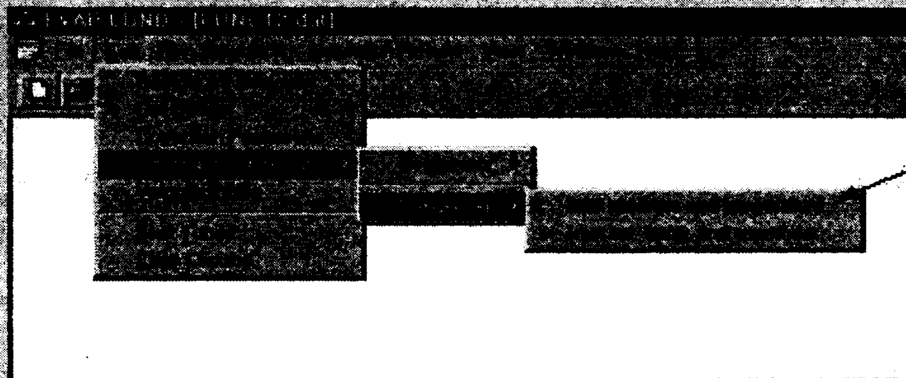
SIMULATION SUMMARY (cont.)

(Complete printout of file st.res)

```
--------------------EVAP5 SIMULATION SUMMARY--------------------

Coil ID: EXAMPLE-5

REFRIGERANT: R22

REFRIGERANT SIDE
Refrigerant mass flow rate:              120.0   [kg/h]
Sensible capacity:                         3.735 [kW]
Latent capacity:                           1.677 [kW]
Total capacity:                            5.413 [kW]
Outlet saturated temp. and superheat:      7.0       6.2   [C]
Inlet and outlet temperatures:             8.7      13.3   [C]
Inlet and outlet pressures:              655.0     622.9   [kPa]
Inlet and outlet qualities:                0.200     1.000

AIR SIDE
Air volumetric flow rate:                 15.0   [m^3/min]
Air temperature distribution [C]:         26.7   21.0   17.8   13.7
Air humidity    distribution [%]:        .5000  .6638  .7411  .8877

CONDITION OF REFRIGERANT LEAVING OUTLET TUBES
Tube     Quality     Temperature     Superheat     Ref. M. Fract.
 #         (-)           (C)            (C)             (-)

 1        0.989          7.6            0.0           0.522
16        1.000         23.0           16.0           0.478

Multiplier for refrig. heat transfer coeff.: 1.00
Multiplier for refrigerant pressure drop..: 1.00
Multiplier for air-side heat trans. coeff.: 1.00
```

CONDENSER OPERATING CONDITIONS

(Two options)



This option allows simulations above the critical point of refrigerant

Both input options include air inlet temperature, pressure, relative humidity, and volumetric flow rate

Execution of COND5 and data analysis is the same as for EVAP5

HOW TO SIMULATE EVAPORATOR ?

(An example using the existing file EXAMPLE-5.dat)

Run Windows Explorer and go to the directory containing EVAP-COND.exe.

Double-click on EVAP-COND.exe to start the program.

Open file EXAMPLE-5.dat to simulate the evaporator. After the file is loaded, you will see a schematic representing a side view of the evaporator. The red circle(s) indicates the inlet tube to the evaporator. The blue circles indicate the outlet tubes. The horizontal line at the bottom of the screen indicates the air velocity profile at the evaporator inlet.

Review Input Data. Click on the *Edit/Coil Design* menu item to review the evaporator design information. You may select either the SI or British system of units for your input data and simulation results.

Click on the *Edit/Operating Conditions/Evaporator/inlet pressure and quality* menu item to review operation conditions. Note, that the loaded option has a mark on the left-hand side. Since EVAP5 simulates performance tube-by-tube from the inlet to outlet, the options that specify any outlet refrigerant parameter involve iterative calls to the option that specifies refrigerant inlet pressure and quality until the target outlet parameters are obtained (e.g., saturation temperature and superheat).

Click on the *Edit/Velocity Profile* menu item to review the air velocity profile. You may use the air mass flow rate specified before or integrate the air velocity profile. In general, the first option is recommended unless very detailed and accurate local measurements of the velocity profile were taken. You may change the air velocity profile using a mouse by clicking the left button.

Run a simulation. Click on the *Run Simulation* menu item and select EVAP5. An MS-DOS window will appear and will give you a message when a simulation run is successfully completed.

Examine local and global simulation results. EVAP5 writes global simulation results to file SI.res (SI system of units) and BT.res (British system of units). The same information is provided in the pull-down menu in the units selected for data input.

HOW TO PREPARE YOUR DATA FILE ?

Start with *Edit/Coil Design* menu item. Input all information.

Select *Edit/Operating Conditions* menu item to input operating conditions data.

Select *Edit/Velocity Profile* to change the velocity profile using a mouse (left button).

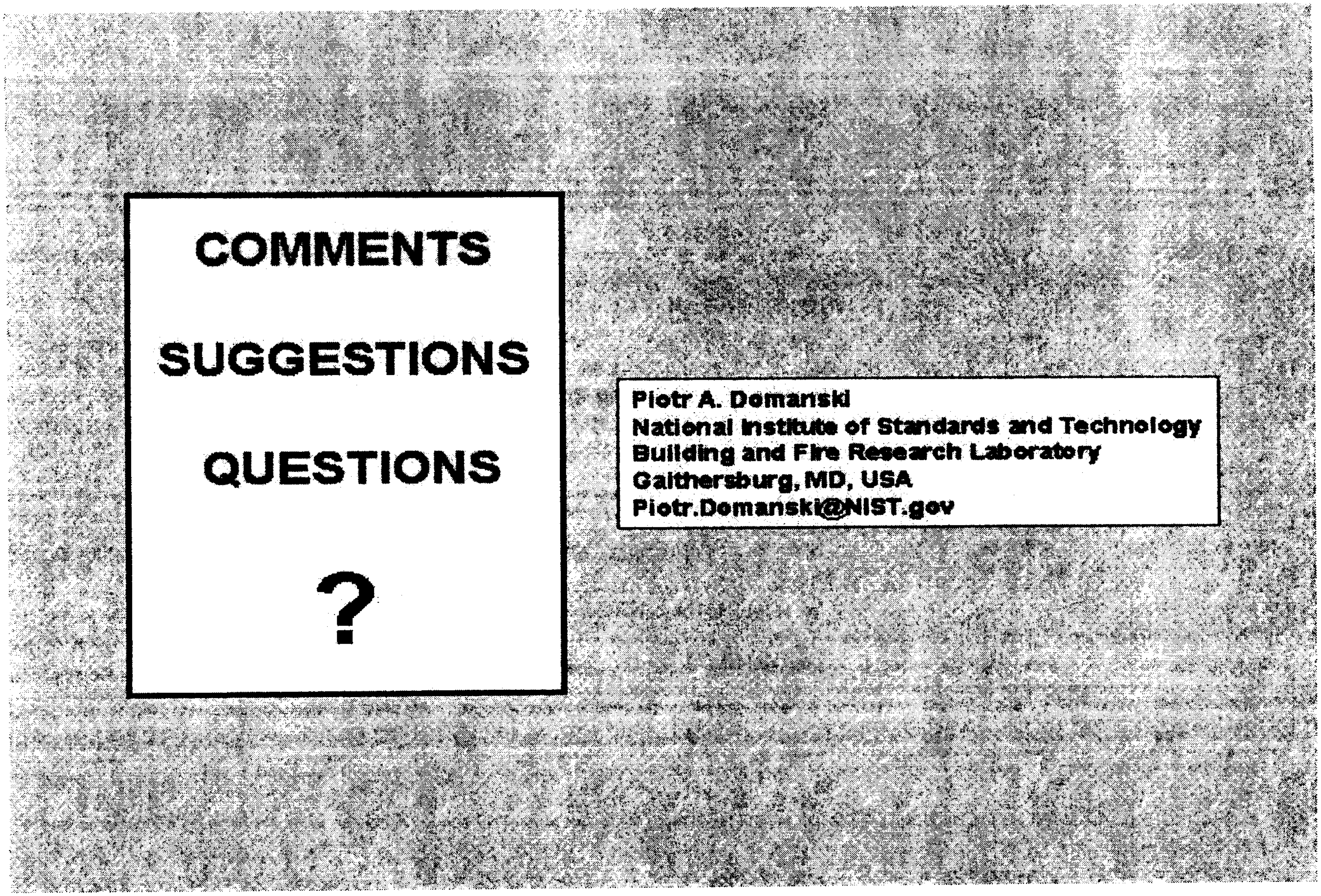
Specify refrigerant circuitry.

- For the heat exchanger working as the evaporator and condenser, start with one of the inlet tubes for the evaporator. The same data file will be used for evaporator and condenser simulations. To draw a return bend, point the mouse on a tube, press the left button, drag the mouse to the next tube, and release. If you want to modify a circuitry, you may delete a part of it from a given tube to the exit tube by pointing the mouse on the given tube and double-clicking the left button.

- For the heat exchanger working exclusively as the condenser, start with one of the outlet tubes and proceed upstream.

CURRENT LIMITATIONS OF EVAP- COND

- Maximum number of tubes in the heat exchanger: 130
- Maximum number of tubes in a depth row: 50
- Maximum number of tube depth rows: 5
- Maximum difference between the number of tubes in different depth rows: 1
- No empty tube locations (no missing tubes in a depth row)
- No merging refrigerant points in the evaporator circuitry; no split circuitry points in the condenser
- Minimum refrigerant temperature in the evaporator: 0 °C


COMMENTS
SUGGESTIONS
QUESTIONS
?
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